TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 27

NEW YORK MEETING, 1905

CHATTANOOGA MEETING, 1906

NEW YORK CITY:

PUBLISHED BY THE SOCIETY

NO. 12 WEST 31ST STREET

1906

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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OFFICERS

1905–1906

FORMING THE STATUTORY COUNCIL

PRESIDENT

FRED. W. TAYLOR . . . . Philadelphia, Pa.

VICE-PRESIDENTS

S. M. VAUCLAIN . . . . Philadelphia, Pa.
H. H. WESTINGHOUSE . . . . Pittsburg, Pa.
GEORGE H. BARRUS . . . . Boston, Mass.
Terms expire at Annual Meeting of 1906

WALTER M. MCFARLAND . . . . Pittsburg, Pa.
EDWARD N. TRUMP . . . . Syracuse, N. Y.
ROBERT C. MCKINNEY . . . . New York, N. Y.
Terms expire at Annual Meeting of 1907

MANAGERS

GEORGE I. ROCKWOOD . . . . Worcester, Mass.
JOHN W. LIEB, JR. . . . . New York, N. Y.
ASA M. MATTICE . . . . Pittsburg, Pa.
Terms expire at Annual Meeting of 1906

GEO. M. BRILL . . . . Chicago, Ill.
FRED. J. MILLER . . . . New York, N. Y.
RICHARD H. RICE . . . . Lynn, Mass.
Terms expire at Annual Meeting of 1907

WALTER LAIDLAW . . . . Cincinnati, O.
FRANK G. TALLMAN . . . . Wilmington, Del.
FREDERICK M. PRESCOTT . . . . Milwaukee, Wis.
Terms expire at Annual Meeting of 1908

TREASURER

WM. H. WILEY . . . . 43–45 East 19th St., New York, N. Y.

SECRETARY

PROF. F. R. HUTTON . . . . 12 West 31st St., New York, N. Y.

HONORARY COUNCILLORS

PAST PRESIDENTS OF THE SOCIETY

THURSTON, R. H.	1880—1882	Died Oct. 25, 1903
LEAVITT, E. D.	1882—1883	Cambridge, Mass.
SWEET, JOHN E.	1883—1884	Syracuse, N. Y.
HOLLOWAY, J. F.	1884—1885	Died Sept. 1, 1896
SELLERS, COLEMAN	1885—1886	Philadelphia, Pa.
BABCOCK, GEORGE H.	1886—1887	Died Dec. 16, 1893
SEE, HORACE	1887—1888	
TOWNE, HENRY R.	1888—1889	New York, N. Y.
SMITH, OBERLIN	1889—1890	Bridgeton, N. J.
HUNT, ROBERT W.	1890—1891	Chicago, Ill.
LORING, CHARLES H.	1891—1892	Brooklyn, N. Y.
COXE, ECKLEY B.	1892—1894	Died May 13, 1895
DAVIS, E. F. C.	1894	Died Aug. 6, 1895
BILLINGS, CHARLES E.*	1895	Hartford, Conn.
FRITZ, JOHN	1895—1896	Bethlehem, Pa.
WARNER, WORCESTER R.	1896—1897	Cleveland, O.
HUNT, CHARLES WALLACE	1897—1898	New York, N. Y.
MELVILLE, GEORGE W.	1898—1899	Philadelphia, Pa.
MORGAN, CHARLES H.	1899—1900	Worcester, Mass.
WELLMAN, S. T.	1900—1901	Cleveland, O.
REYNOLDS, EDWIN	1901—1902	Milwaukee, Wis.
DODGE, JAMES M.	1902—1903	Philadelphia, Pa.
SWASEY, AMBROSE	1903—1904	Cleveland, O.
FREEMAN, JOHN R.	1904—1905	Providence, R. I.

Note.—According to the Constitution, Article C 27, the five surviving Past Presidents who last held the office are members of the Council, with all the rights, privileges and duties of other members.

* Unexpired term of E. F. C. Davis.

NOTE

The considerable bulk of the volume of Transactions has induced the Publication Committee to direct the insertion of a summary of the Society membership in place of the complete list of members which was published in the earlier volumes. The summary attaching to this issue is that which appears in the List of Members (Pocket Edition), issued July, 1906.

FOREIGN COUNTRIES

	Membership		Membership
Africa	20	India	2
Australia	7	Japan	6
Belgium	6	Mexico	8
Canada	29	Norway	1
China	1	Panama	2
Cuba	5	Russia	5
France	8	South America	8
Germany	10	Sweden	3
Great Britain (England)	49	Switzerland	2
Great Britain (Scotland)	4	Trinidad, B. W. I.	1
Holland	1		

Total 178

UNITED STATES

	Membership		Membership
Alabama	10	Nevada	1
Alaska	1	New Hampshire	16
Arkansas	3	New Jersey	154
California	45	New Mexico	1
Colorado	21	New York	868
Connecticut	123	North Carolina	7
Delaware	14	North Dakota	1
District Columbia	25	Ohio	221
Georgia	10	Oklahoma	1
Hawaiian Islands	2	Oregon	5
Illinois	198	Pennsylvania	387
Indiana	39	Porto Rico	1
Iowa	7	Rhode Island	51
Kansas	8	South Carolina	4
Kentucky	6	Tennessee	6
Louisiana	18	Texas	7
Maine	16	Utah	4
Maryland	33	Vermont	12
Massachusetts	264	Virginia	30
Michigan	76	Washington	5
Minnesota	11	West Virginia	10
Missouri	45	Wisconsin	1
Montana	11	Wyoming	1
Nebraska	6		

Total 2,856

GEOGRAPHICAL SUMMARY

Total Foreign Membership	178
Total Membership in United States	2,856
* Present Address Unknown	6
Total Membership	3,040

SUMMARY BY GRADES

Honorary Members	19
Members	2,058
Associates	273
Junior Members	690
Total Membership	3,040
† Life Members	

* These are:

Fawcett, Wallace H.
Grimshaw, Frederick G.
Hillard, Charles J.
Mayer, L. G. C.
Porter, Arthur T.
Stiles, Norman C.

If any member knows the present addresses of any of these members, he will confer a favor by advising the Secretary.

† These Life Members are included in the total membership above, in the class to which they belong.

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TO THE

CONSTITUTION, BY-LAWS AND RULES.

AMERICAN SOCIETY MECHANICAL ENGINEERS.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

CONSTITUTION.

NAME, OBJECT AND GOVERNMENT.

C 1. The title of this Society is "The American Society of Mechanical Engineers."

C 2. The object of the Society is to promote the Arts and Sciences connected with Engineering and Mechanical Construction. The principal means for this purpose shall be the holding of meetings for the reading and discussion of professional papers, and for social intercourse; the publication and distribution of its papers and discussions; and the maintenance of an Engineering Library.

C 3. The Society shall be governed by this Constitution, and by By-Laws and Rules in harmony therewith.

C 4. The Society was organized as a Corporation under the laws of the State of New York, April 7, 1880. Its offices shall be located in the City of New York.

MEMBERSHIP.

C 5. Persons connected with the Arts and Sciences relating to Engineering or Mechanical Construction may be eligible for admission into the Society.

C 6. The membership of the Society shall consist of Honorary Members, Members, Associates and Juniors. Honorary Members, Members and Associates are entitled to vote and to hold office. Juniors shall not be entitled to vote nor to be officers of the Society, but shall be entitled to the other privileges of membership.

C 7. Honorary Members, Members and Associates are en-

titled to vote on all questions before any meeting of the Society, in person or by proxy, given to a voting member. A proxy shall not be valid for a greater time than six months.

C 8. Honorary Members shall be persons of acknowledged professional eminence, and their number shall not exceed twenty-five at any time.

C 9. A Member must have been so connected with Engineering as to be competent, as a designer or as a constructor, to take responsible charge of work in his branch of Engineering, or he must have served as a teacher of Engineering for more than five years. A Member shall be thirty years of age or over.

C 10. An Associate must either have the other qualifications of a Member or be so connected with Engineering as to be competent to take charge of engineering work, or to co-operate with Engineers. An Associate shall be twenty-six years of age or over.

C 11. A Junior must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school. A Junior shall be twenty-one years of age or over.

C 12. The rights and privileges of every Honorary Member, Member, Associate and Junior shall be personal to himself, and shall not be transferable or transmissible by his own act or by operation of law.

ADMISSION.

C 13. Honorary Members shall be nominated by at least ten members of the Society. The grounds upon which the nomination is made, shall be presented to the Council in writing.

C 14. All applications for membership to the grades of Member, Associate or Junior shall be presented to the Council, which shall consider and act upon each application, assigning each approved applicant to the grade of membership to which, in the judgment of the Council, his qualifications entitle him. The name of each candidate thus approved by the Council, shall, unless objection is made by the applicant, be submitted to the voting membership for election, by means of a letter-ballot.

C 15. Associates or Juniors desiring to change their grade of membership shall make application to the Council in the same manner as is required in the case of a new applicant.

C 16. Election to membership shall be by a sealed letter-ballot as the By-Laws shall provide. Adverse votes to the number of two per cent. of the votes cast shall be required to defeat the election of an applicant for any grade of membership. The Council, may in its discretion, order a second ballot upon a defeated applicant, in which case adverse votes to the number of four per cent. of the votes cast, shall be required to defeat the election.

C 17. The election of Honorary Members shall be by a vote of the Council taken by letter-ballot, as provided in the By-Laws. One dissenting vote shall defeat such election.

C 18. Each person elected, excepting Honorary Members, shall subscribe to this Constitution, and shall pay the initiation fee before he can be entitled to the rights and privileges of membership. If such person does not comply with this requirement within six months after notice of his election, he will be deemed to have declined election. The Council may, thereupon, declare his election void.

INITIATION FEES AND DUES.

C 19. The initiation fee for membership in each grade shall be as follows:

For Member................Twenty-five Dollars,
For Associate...............Twenty-five Dollars,
For Junior..................Fifteen Dollars.

C 20. A Junior, on promotion to any other grade of membership, shall pay an additional fee of Ten Dollars.

C 21. The annual dues for membership in each grade shall be as follows:

For Member.....................Fifteen Dollars,
For Associate....................Fifteen Dollars,
For Junior................Ten Dollars for the first six years of his membership and thereafter the same as for an Associate.

C 22. The Council may in its discretion, permit any Member or Associate to become a Life Member in the same grade, by the payment at one time of an amount sufficient to purchase from the Equitable Life Assurance Society of New York, an annuity on the life of a person of the age of the applicant equal to the

annual dues in his grade. Such Life Member shall not be liable thereafter for annual dues.

C 23. The Council shall have the power, by letter-ballot, to admit to Life Membership, without the payment of a life membership fee, any person who, for a long term of years, has been a Member or an Associate when, for special reasons, such procedure would, in its judgment, promote the best interests of the Society, provided that notice of such proposed action shall have been given at a previous meeting of the Council. One dissenting vote shall defeat such admission.

SUSPENSIONS AND EXPULSIONS.

C 24. Any Member, Associate or Junior who shall leave his annual dues unpaid for one year, shall not receive the volume of *Transactions* until such arrears are paid. Any Member, Associate or Junior who shall leave his dues unpaid for two years, shall, in the discretion of the Council, have his name stricken from the roll of membership, and shall cease to have any further rights as such.

C 25. The Council may refuse to receive the dues of any member of any grade, who shall have been adjudged by the Council to have violated the Constitution or By-Laws of the Society, or who, in the opinion of the Council by a two-thirds vote, shall have been guilty of conduct rendering him unfit to continue in its membership; and the Council may expel such person and remove his name from the list of members.

THE COUNCIL.

C 26. The affairs of the Society shall be managed by a Board of Directors chosen from among its Members and Associates, which shall be styled "The Council." The Council shall consist of the President of the Society, who shall be the presiding officer, six Vice-Presidents, nine Managers, the Treasurer and five Past Presidents. Five members of the Council shall constitute a quorum for the transaction of business. The Secretary may take part in the deliberations of the Council, but shall not have a vote therein. The Chairman of the Finance Committee shall attend the meetings of the Council and take part in the discussion of financial questions but shall not have a vote.

C 27. The five surviving Past Presidents who last held the office shall be members of the Council with all the rights, privileges and duties of the other members of the Council.

C 28. The Council thus constituted shall be the legal Trustee of the Society. All gifts or bequests not designated for a specific purpose shall be invested by the Council, and only the income therefrom may be used for current expenses.

C 29. Should a vacancy occur in the Council, or in any elective office except the presidency, through death, resignation or other cause, the Council may elect a Member or Associate to fill the vacancy until the next annual election.

C 30. The Council shall regulate its own proceedings, and may by resolution delegate specific powers to an Executive Committee or to any one or more members of the Council. No act of the Executive Committee or of a delegate shall be binding until it has been approved by a resolution of the Council.

C 31. The Council shall present at the Annual Meeting of the Society a report verified by the President or Treasurer or by a majority of the members of the Council, showing the whole amount of real and personal property owned by the Society, where located, and where and how invested, and the amount and nature of the property acquired during the year immediately preceding the date of the report, and the manner of the acquisition; the amount applied, appropriated or expended during the year immediately preceding such date, and the purposes, objects or persons to or for which such applications, appropriations or expenditures have been made; also the names and places of residence of the persons who have been admitted to membership in the Society during the last year, which report shall be filed with the records of the Society, and an abstract thereof shall be entered in the minutes of the proceedings of the Annual Meeting.

C 32. An act of the Council, which shall have received the expressed or the implied sanction of the membership at the next subsequent meeting of the Society, shall be deemed to be the act of the Society, and shall not afterwards be impeached by any member.

C 33. The Council may, by a two-thirds vote of the members present, declare any elective office vacant, on the failure of its incumbent for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office, and shall thereupon appoint a Member or Associate to fill the vacancy

until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office.

OFFICERS.

C 34. At each Annual Meeting there shall be elected from among the Members and Associates:

A President to hold office for one year.

Three Vice-Presidents, each to hold office for two years.

Three Managers, each to hold office for three years.

A Treasurer to hold office for one year.

C 35. The election of officers shall be by sealed letter-ballot, as the By-Laws shall provide.

C 36. The term of all elective officers shall begin on the adjournment of the Annual Meeting of the Society. Officers shall continue in their respective offices until their successors have been elected and have accepted their offices.

C 37. A President, Vice-President or Manager shall not be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

C 38. The Council, at its first meeting after the Annual Meeting of the Society, shall appoint a person of the grade of Member to serve as Secretary of the Society for one year, subject to removal for cause by an affirmative vote of fifteen members of the Council, at any time after one month's written notice has been given him to show cause why he should not be removed, and he has been heard in his own defense, if he so desires. The Secretary shall receive a salary which shall be fixed by the Council at the time of his appointment.

C 39. The President, Secretary and Treasurer shall perform the duties legally or customarily attaching to their respective offices under the Laws of the State of New York, and such other duties as may be required of them by the Council.

C 40. A vacancy in the office of President shall be filled by the Vice-President, who is senior by age.

MEETINGS.

C 41. The Society shall hold two meetings in each year. The Annual Meeting shall begin in New York City on the first Tuesday in December, and a Semi-Annual Meeting shall be held at

such time and place as the Council may appoint. Fifty Members and Associates shall constitute a quorum for the transaction of business.

C 42. Special meetings of the Society may be called at any time at the discretion of the Council, or shall be called by the President upon the written request of fifty members entitled to vote, the notices for such meetings to state the business for which such meeting is called, and no other business shall be entertained or transacted at that meeting.

C 43. Any appropriation recommended by the Society at a meeting shall not take effect until it has been approved by the Council.

C 44. Every question which shall come before a meeting of the Society or of the Council or a Committee, shall be decided by a majority of the votes cast, unless otherwise provided in this Constitution or the By-Laws, or the Laws of the State of New York. The Council may order the submission of any question to the membership for discussion by letter-ballot. Any meeting of the Society at which a quorum is present, may order the submission of any question to the membership for discussion by letter-ballot.

STANDING COMMITTEES.

C 45. The Standing Committees of the Society to be appointed by the President shall be:

Finance Committee,
Committee on Meetings,
Publication Committee,
Membership Committee,
Library Committee,
House Committee.

C 46. There shall be a John Fritz Medal Committee of three members appointed as provided in the By-Laws.

C 47. The Annual Committees shall be:

An Executive Committee, appointed by the Council.

A Nominating Committee, appointed by the President.

Tellers as required by the By-Laws, appointed by the President.

C 48. Special Nominating Committee:

Twenty or more members entitled to vote may constitute

themselves a Special Nominating Committee, with the same powers as the Annual Nominating Committee.

C 49. Professional Committees:

The Council shall have power to appoint, upon a recommendation of the Society at a general meeting, or upon its own initiative, such Professional Committees as it may deem desirable, to investigate, consider and report upon subjects of engineering interest. Reports of such committees may be accepted by the Society and printed in the *Transactions*, but shall not be approved or adopted as the action of the Society. Any proposed expenses of such committees must be authorized by the Council before they are incurred.

C 50. Each Committee shall perform the duties required of it in the By-Laws, or assigned to it by the Council. The Secretary of the Society shall be the Secretary of each of the Standing Committees.

C 51. The Council may at any time, in its own discretion, remove any or all members of any Committee, except a Nominating Committee; and the vacancy, arising from this or from any other cause, shall be filled by appointment by the President, except a vacancy in the Executive Committee, which shall be filled by the Council.

SECTIONS OF THE SOCIETY.

C 52. The Council may, in its discretion, authorize the organization of sections or groups of any or all grades of membership, for professional or scientific purposes which are in harmony with the Constitution and By-Laws of this Society. Such sections or groups may, in the discretion of the Council, be geographical or professional, and shall have such powers, and act under such rules and regulations as the Council may from time to time prescribe.

TRANSACTIONS.

C 53. The official record of technical papers and discussion, shall be known as the *Transactions* of the Society, and shall be published under the direction of the Council. There may be included therein, the annual report of the Council, reports of Committees, and business records of the Society.

C 54. The Society shall claim no exclusive copyright to any papers read at its meetings, or any reports or discussions thereon, except in the matter of their official publication under the Society's imprint as its *Transactions*. The policy of the Society shall be to give the professional and scientific papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging Engineering progress and extending the professional reputation of its members.

C 55. The Society shall not be responsible for statements or opinions advanced in papers or in discussions at its meetings. Matters relating to politics or purely to trade shall not be discussed at a meeting of the Society, nor be included in the *Transactions*.

C 56. The Society shall not approve or adopt any standard or formula, or approve any engineering or commercial enterprise. It shall not allow its imprint or name to be used in any commercial work or business.

AMENDMENTS TO THE CONSTITUTION.

C 57. At any semi-annual meeting of the Society any member may propose in writing an amendment to this Constitution. Such proposed amendment shall not be voted on at that meeting, but shall be open to discussion and to such modification as may be accepted by the proposer. The proposed amendment shall be mailed in printed form by the Secretary to each member of the Society entitled to vote, at least sixty days previous to the next annual meeting, accompanied by comment by the Council, if it so elects. At that annual meeting such proposed amendment shall be presented for discussion and final amendment, and shall subsequently be submitted to all members entitled to vote, provided that twenty votes are cast in favor of such submission. The final vote on adoption shall be by sealed letter-ballot, closing at twelve o'clock noon on the first Monday of March following.

C 58. The letter-ballot, accompanied by the text of the proposed amendment, shall be mailed by the Secretary to each member of the Society entitled to vote at least thirty days previous to the closure of the voting. The ballots shall be voted, canvassed and announced as provided in the By-Laws. The adoption of the amendment shall be decided by a majority of the

votes cast. An amendment shall take effect on the announcement of its adoption by the Presiding Officer of the semi-annual meeting next following the closure of the vote.

AMENDMENTS TO BY-LAWS AND RULES.

C 59. For the further ordering of the affairs of the Society, the Council may, by a two-third vote of its members present, amend the By-Laws in harmony with this Constitution, provided that a written notice of such proposed amendment shall have been given at the previous regular meeting of the Council; and provided further that the Secretary shall have mailed to each member of the Council a copy of such proposed amendment, at least thirty days in advance of the meeting of the Council at which action is to be taken. The amendment shall take effect immediately on its passage by the Council. The Secretary shall at once mail a copy of such amendment to the members of all grades.

C 60. The Council may, by a majority vote of the members present at any meeting, establish, amend or annul Rules for the conduct of the business affairs of the Society; for the ordering and conduct of its professional or business meetings; and for guidance of its committees in their work and reports; provided that such Rules are in harmony with the Constitution and By-Laws of the Society.

CONSTITUTION GOES INTO EFFECT.

C 61. This Constitution shall supersede all previous Rules of the Society, and shall go into effect on the announcement by the Presiding Officer of its adoption.

BY-LAWS.

CANDIDATES FOR MEMBERSHIP.

B 1. A candidate for admission to the Society as a Member or as an Associate must make application on a form approved by the Council, upon which he shall write a statement giving a complete account of his qualifications and engineering experience, and an agreement that he will, if elected, conform to the Con-

stitution, By-Laws and Rules of the Society. He must refer to at least five Members or Associates to whom he is personally known.

B 2. Applications for membership from Engineers who are not resident in the United States or Canada, and who may be so situated as not to be personally known to five Members of the Society, as required in the foregoing paragraph, may be recommended for ballot by five members of the Council, after sufficient evidence has been secured to show that in their opinion the applicant is worthy of admission to the grade which he seeks.

B 3. A candidate for admission to the Society as a Junior must make application in the same manner as provided for Members, except that he must refer to not less than three Members or Associates to whom he is personally known.

B 4. References shall not be required of candidates for Honorary Membership.

B 5. The references for each candidate for admission to the Society shall be requested to make a confidential communication to the Membership Committee, setting forth in detail such information, personally known to referee, as shall enable the Council to arrive at a proper estimate of the eligibility of the candidate for admission to the Society.

ELECTION OF MEMBERS.

B 6. The Secretary shall mail to each member entitled to vote, at least thirty days in advance of each annual or semi-annual meeting, a ballot stating the names and the respective grades of the candidates for membership in the Society which have been approved by the Council, and the time of the closure of voting. The voter shall prepare his ballot by crossing out the names of candidates rejected by him, and shall enclose said ballot in a sealed blank ballot envelope which he shall then enclose in a second sealed outer envelope on which he shall, for identification, write his name in ink. The ballot thus prepared and enclosed shall be mailed or delivered unopened to the Tellers of Election. The Secretary shall certify to the competency, and the signature of all voters. On the closure of voting, the Tellers of Election shall first open and destroy the outer envelopes, and shall then canvass the ballots, and certify the result to the meeting of the Society.

B 7. The Tellers shall not receive any ballot after the stated time of the closure of voting. A ballot without the endorsement of the voter, written in ink on the outer envelope, is defective, and shall be rejected by the Tellers of Election.

B 8. The names of those persons elected to membership, with their respective grades, shall be embodied in a written report, signed by the Tellers, and presented to the next meeting of the Society. The President shall then declare them duly elected to membership in the Society. The Tellers may, through the Secretary, in advance of any meeting advise each candidate of the result of the canvass of the votes in his case. The names of applicants who are not elected shall neither be announced nor recorded in the *Transactions*.

B 9. The endorsers of an applicant who has not been elected, may, with his consent, present to the Council a written request for a re-submission of his name to ballot. The Council may, in its discretion, by a three-fourths vote of the members present, order the name of the applicant placed on the next ballot for members.

B 10. Election to Honorary Membership shall be by letter-ballot of the Council. A notice of such proposed election shall be mailed by the Secretary to each member of the Council at least sixty days in advance of the date set for the closure of such election.

B 11. Each person elected to membership, except an Honorary Member, must subscribe to the Constitution, By-Laws and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society.

ELECTION OF OFFICERS.

B 12. The Secretary shall mail to each member entitled to vote, at least thirty days before the Annual Meeting, the names of the candidates for office proposed for election by the Nominating Committee.

B 13. The names of the candidates proposed by the Nominating Committee or Committees, and the respective offices for which they are candidates, shall be printed in separate lists on the same ballot sheet, each list of candidates to be printed under the names of the members of the particular committee which proposed it.

B 14. The name of any candidate on the ballot may be erased, and the name of any person qualified to hold the office written in its stead. The ballot thus prepared must be voted and canvassed in the same manner as for the election of members.

B 15. At the first session of the Annual Meeting, the Tellers of Election of Officers shall canvass the votes cast for the officers of the Society in the manner prescribed for the election of members, and immediately report the result of the canvass to the meeting. The President shall then announce the candidates having the greatest number of votes for their respective offices, and declare them elected for the ensuing year.

B 16. In case of a tie in the vote for any officer, the President or, in his absence, the Presiding Officer shall cast the deciding vote.

B 17. A ballot which contains more names marked by a cross on it than there are officers to be elected, is thereby defective, and shall be rejected by the Tellers.

FEES AND DUES.

B 18. The initiation fee and annual dues of the first year shall be due and payable on notice of election to membership, and upon that payment the member will be entitled to the *Transactions* for the year. Thereafter the annual dues shall be due and payable on the first day of October in each year.

B 19. A member in arrears for one year shall not be entitled to vote until such arrears have been paid. Should the right to vote be questioned, the books of the Society shall be conclusive evidence.

B 20. The Secretary shall present to the Council the name of any Member, Associate or Junior in arrears for more than one year, and such member shall not receive the *Transactions* until such arrears are fully paid. A person dropped from the rolls for non-payment of dues may, in the discretion of the Council, be restored to the privileges of membership, upon payment of all arrears.

FINANCIAL ADMINISTRATION.

B 21. The Council at its first meeting in each fiscal year, shall consider the recommendations of the Finance Committee

concerning the expenditure necessary for the work of the Society during that year. The apportioning of the work of the Society among the various Standing and other Committees shall be on a basis approved by the Council and in harmony with the Constitution and By-Laws. The appropriations approved by the Council, or so much thereof as may be required for the work of the Society, shall be expended by the various Committees of the Society, and all bills against the Society for such expenditure shall be certified by the Committee making the expenditure and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employee of the Society except upon bills duly audited by the Finance Committee, or by resolution of the Council.

COMMITTEES.

B 22. The President within one month after the Annual Meeting shall fill all vacancies in the Standing Committees by appointment from the membership of the Society.

Each of the Standing and the Annual Committees, shall, at their first meeting after the Annual Meeting, elect a Chairman to serve for one year. The President shall appoint the Chairman of each Professional Committee. A member of a Standing Committee whose term of office has expired, shall continue to serve until his successor shall have been appointed.

FINANCE COMMITTEE.

B 23. The Finance Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. This Committee shall, in the discretion of the Council, have a supervision of the financial affairs of the Society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent public accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for presentation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate of the prob-

able income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and, when called upon by the Council, advise upon financial questions.

COMMITTEE ON MEETINGS.

B 24. The Committee on Meetings shall consist of five persons who may be members of any grade. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Committee to procure professional papers, to pass upon their suitability for presentation, and to suggest topical subjects for discussion at the meetings. The Committee may refer any paper presented to the Society to a person or persons, especially qualified by theoretical knowledge or practical experience, for their suggestions or opinions as to the suitability of the paper for presentation. Papers from non-members shall not be accepted except by unanimous vote of the Committee.

The Committee shall arrange the programme of each meeting of the Society, and shall have general charge of the entertainments to be provided for the members and guests at each meeting. It shall prohibit the distribution or exhibition at the headquarters or at the meeting places of the Society of all advertising circulars, pamphlets or samples of commercial apparatus or machinery. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

PUBLICATION COMMITTEE.

B 25. The Publication Committee shall consist of five Members or Associates. The term of office of one member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers or discussions, or parts of the same, shall be printed in the *Transactions* of the Society. The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

MEMBERSHIP COMMITTEE.

B 26. The Membership Committee shall consist of five Members or Associates. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of this Committee:

To meet monthly to receive and scrutinize all applications for membership to the Society.

To send to each voting member the name, qualifications, engineering experience and references of each applicant, together with extracts from the Constitution and By-Laws relating to membership.

To seek further information as to the qualifications of an applicant, whose evidence of eligibility is not clear to the Committee.

To report to each session of the Council the names of all applicants under consideration together with the action of the Committee on each.

The Committee shall at once destroy all correspondence in relation to each applicant when his name has been placed on the ballot by order of the Council, or upon the withdrawal of the application.

LIBRARY COMMITTEE.

B 27. The Library Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Library Committee to take charge of the Library of the Society, the historical relics, the paintings and objects of art, and to recommend to the Council suitable regulations for their care and use. At the end of each fiscal year, the Committee shall deliver to the Secretary, a detailed report of its work.

HOUSE COMMITTEE.

B 28. The House Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the House Committee to have the care, management and maintenance of the house of the Society and its furnishings. They may make rules for the care and the use

of the Society House, subject to the approval of the Council. At the end of each fiscal year, the Committee shall deliver to the Secretary a detailed report of its work.

EXECUTIVE COMMITTEE.

B 29. The Council shall appoint from its members an Executive Committee to act for the Council during the interval between its sessions. The Committee shall make a report of its acts to each session of the Council for approval. The Secretary may take part in the deliberations of the Executive Committee, but shall not have a vote therein.

NOMINATING COMMITTEES.

B 30. A Nominating Committee of five Members, not members of the Council, shall be appointed by the President within three months after he assumes office. It shall be the duty of this Committee to send to the Secretary on or before October first the names of consenting nominees for the elective offices next falling vacant under the Constitution. Upon the request of any Member or Associate, the Secretary shall furnish to the applicant the names of such nominees.

B 31. A special Nominating Committee if organized, shall, on or before October twentieth, present to the Secretary the names of the candidates nominated by it for the elective offices next falling vacant under the Constitution, together with the written consent of each.

JOHN FRITZ MEDAL COMMITTEE.

B 32. The John Fritz Medal Committee shall consist of three persons of the grade of Member, to be appointed by the Council. The term of office of one member of this Committee shall expire at the end of each annual meeting. The duty of this Committee shall be to represent the Society in the Board of Trustees of the John Fritz Medal Fund Corporation.

REPRESENTATIVE DELEGATES.

B 33. The Council may in its discretion appoint a member or members of the Society or other person or persons to repre-

sent it at meetings of Societies of kindred aim or at public functions. Such delegates shall be designated as "Honorary Vice-Presidents," and their duties shall terminate with the occasion for which they were appointed.

TELLERS.

B 34. The Presiding Officer shall, at the first session of the Annual Meeting, appoint three Tellers of Election of officers, whose duties shall be to canvass the votes cast, and report the result to the meeting. Their term of office shall terminate when their report of the canvass is presented to the meeting.

B 35. The President within one month after assuming office shall appoint three Tellers of Election of members to serve for one year, whose duties shall be to canvass the votes cast for members during the year, and to certify the same to the President. They shall notify candidates through the Secretary of the result of such election.

B 36. The President shall appoint three Tellers to canvass any letter-ballots which shall be ordered by the Council or by the Society.

MEETINGS.

B 37. The meetings of the Society shall continue from day to day as the meeting may decide. The business session of the Annual Meeting shall be held on Wednesday following the first Tuesday of December. The professional sessions for the reading of papers shall be held at such times and places as the meeting may appoint. Notices of all meetings of the Society shall be mailed by the Secretary to members of all grades not less than thirty days before the date of such meeting.

SECRETARY.

B 38. The Secretary of the Society shall be the Secretary to the Council and also to each of the Standing Committees.

The Secretary shall, under the supervision of the Finance Committee, have charge of the Books of Account of the Society.

He shall make and collect all bills against members or others.

He shall have charge of all bills against the Society, shall

keep an account of the same, and shall present them in proper form to the Finance Committee for audit.

All funds received by any person for the Society, shall be delivered to the Secretary. He shall immediately enter them in the Books of Account, and shall immediately deposit such funds as he receives, to the credit of the Society, in a Bank to be designated by the Council.

TREASURER.

B 39. The Treasurer shall make payments only on the audit of the Finance Committee, or upon the direction of the Council, by resolution of that body. He shall furnish a bond for the faithful performance of his duties to such amount as the Council may require, such bond to be procured from an incorporated Guarantee Company, at the expense of the Society.

TITLES, EMBLEMS, CERTIFICATE.

B 40. Each Member and Associate shall, subject to such rules as the Council may establish, be entitled on request, to a certificate of membership, signed by the President and Secretary of the Society. Every such certificate shall remain the property of the Society, and shall be returned to it on demand of the Council.

B 41. Each proxy authorizing a person to vote for an absent member, shall be signed by such absent member, with an attesting witness, and be submitted to the Secretary for verification of the member's right to vote at the meeting at which the right is to be exercised.

B 42. The emblem of each grade of membership approved by the Council shall be worn by those only who belong to that grade. The official stationary shall be used only by Officers and Committees of the Society.

B 43. The abbreviation of the titles of the various grades of membership approved by the Society are as follows:

For Honorary Members, . .	Hon. Mem. Am. Soc. M. E.
For Members, ...	Mem. Am. Soc. M. E.
For Associates, ...	Assoc. Am. Soc. M. E.
For Juniors, ...	Jun. Am. Soc. M. E.

RULES.

R 1. The Secretary's office shall be open on business days from 9 A.M. to 5.30 P.M. During the Annual Meeting, the office shall be open from 9 A.M. to 10 P.M. A register shall be kept for each regular meeting, to record the attendance of members and guests.

R 2. The Secretary shall provide a numbered badge or pin for each member or guest attending the regular meetings, the number on the badges to correspond with the member's of guest's number on the register.

R 3. The Secretary shall at each regular meeting of the Society distribute at the headquarters a printed list of the names registered at the meeting.

R 4. Copies of papers to be read and discussed at any meeting shall be sent to each member thirty days in advance of that meeting. A paper received too late for such distribution shall only be accepted for presentation at that meeting by unanimous consent of the Committee on Meetings. A blank shall accompany the papers by which a member may signify his intention to discuss any of the papers, and priority in debate shall be given in the order of the receipt by the Secretary of such notification.

R 5. At professional sessions, each paper shall be read by abstract only, ten minutes being allowed to the author for the presentation, unless otherwise ordered by the meeting.

R 6. A member who has given notice of his intention to discuss a paper, and shall have reduced his discussion to writing, shall be entitled to ten minutes for its presentation.

R 7. Each speaker shall be limited to five minutes in the oral discussion of a paper, unless the time should be extended by unanimous consent. A member who has once had the floor cannot claim it again until all the others have been heard who desire to speak on that paper. Authors may have five minutes to close the discussion on the paper.

R 8. Members unable to attend the meeting may send a discussion of any paper in writing, to be presented by the Secretary.

R 9. The Committee on Meetings shall deliver to the Secretary such papers as they recommend for presentation to the professional meetings of the Society.

R 10. The Secretary shall have sole possession of papers and illustrations between the time of their approval by the Committee on Meetings, and their presentation to the professional session of the Society.

R 11. After the presentation and discussion of a paper, a copy of both shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to each member who presented it, with the request that he correct errors or omissions, and return the same promptly to the Secretary.

R 12. Members may order reprints of papers at a price sufficient to cover the cost to the Society, provided that said copies are not for sale.

R 13. The Secretary may furnish to the author twenty copies of his paper without charge. He may also furnish to the technical press such papers in advance of the meeting as they may wish to publish after presentation to the meeting of Society.

R. 14. The entertainments to be provided for the members and guests at any meeting of this Society in any city shall be in charge of a Local Committee, subject, however, to the general approval of the Committee on Meetings.

R 15. A member may invite a non-member to the professional sessions of the meeting, but the guest shall not take part in the proceedings without an invitation from the Presiding Officer. Invitations to guests of members for the entertainments provided for the Society shall be in the discretion of the Local Committee.

R 16. The Society House shall be open at all hours for access to members. The Library shall be open on all week days between the hours of 10 o'clock A.M. and 10 o'clock P.M. It shall be conducted as a Free Public Reference Library of Engineering and the Allied Arts and Sciences.

R 17. Juniors who were elected to membership in the Society six years or more previous to the adoption of this Constitution, shall pay the same dues as an Associate, beginning with the fiscal year which opens after such adoption. Juniors, who have been elected less than six years before that date, shall pay the dues of an Associate on the expiration of six years after their election.

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NEW YORK MEETING

HELD ON DECEMBER 5, 6, 7 AND 8, 1905,
BEING THE ANNUAL MEETING AND THE
FIFTY-SECOND MEETING OF THE SOCIETY

No. 1095

PROCEEDINGS OF THE NEW YORK MEETING

The twenty-sixth annual meeting of the Society was held in the city of New York on the first Tuesday of December, 1905, as required by the Constitution of the Society. The increasing growth of the Society membership and the attendant increase in the numbers present at its annual meeting, has made it uncomfortable in the auditorium of the Society and its parlors, so that no attempt was made to have the convention center at the Society House at No. 12 West Thirty-first Street.

The Committee on Meetings therefore were greatly pleased to accept a courteous invitation from the officers of the New York Edison Company for the Society to make use of its auditorium and reading room as meeting place and headquarters for the convention. These rooms are located at No. 44 West Twenty-seventh Street on the third and fourth stories of the converting station for that district. The auditorium will hold comfortably over four hundred, and the room below was a most convenient and comfortable headquarters' area. All the details of registration and serving of the luncheons were provided for on the third floor and the professional sessions were on the fourth floor with the exception of the session on Wednesday morning. The auditorium was most attractively lighted and decorated with palms and festoons of greenery.

At the side of the platform the members of the engineering and executive staff of the New York Edison Company, who are members of the Society, had united in having prepared and presented to the Society a monogram in colored electric lights, the product of one of the most successful firms engaged in this form of construction.

FIRST SESSION. TUESDAY EVENING, DECEMBER 5TH.

The opening session was held on Tuesday evening, December 5th, and was devoted to the address of Mr. John R. Freeman, President of the Society, in which he discussed with great fulness the Relations of the Engineer to the Problem of Fire Protection in Theaters. Mr. Freeman's address appears as one of the papers of the meeting.

At the close of the President's address the Secretary made some announcements with respect to the business before the meeting during the convention period, and notified the members of the courteous action of the Engineers' Club of Central Pennsylvania at Harrisburg, in inviting members of the Society to be its guests in the attractive club house in that city when their business should bring them within reach of it.

The invitation from the American Society of Civil Engineers was also presented, whereby the Board of Direction of that body invited members of the Society to attend the professional meetings as they should occur during the winter, and to use the library of the Society under the same privileges as were extended to their own members.

A light collation was served at the end of the meeting.

SECOND SESSION. WEDNESDAY MORNING, DECEMBER 6TH, 10.30 O'CLOCK.

The second session of the annual meeting is by appointment of the Constitution the Business Session. It was held in the dining saloon of the steamship Amerika, of the Hamburg-American Line, who had invited the Society to be the guests of the Company and inspect the engineering and other features of the steamer as it lay at its pier. The Society was welcomed by Mr. Emil Boas, General Manager of the New York office of the Company. His address was historical, showing the development in trans-Atlantic transportation in fifty years, and particularly in the interval spanned by the preceding twenty years.

During and after the session the opportunity to inspect the ship and its motive power department, were freely availed of by a very large number.

The registration of members and their guests in attendance at the annual meeting was far in excess of any previous meeting of the Society. There were over 1,300 enrolled, of which 700 were members and 600 guests. The presence of this large number added greatly to the interest and social opportunity during the convention days.

The first order of business of the meeting was the presentation of the Report of the Council, which had already been printed and distributed by mail to all members, and was presented by the Secretary for record, as follows:

ANNUAL REPORT OF THE COUNCIL.

Under the provisions of the Constitution the Council of the Society presents to the annual meeting a report of the business which has been under consideration during the year, together with the various reports from standing committees.

The amount of real and personal property owned by the Society and the property acquired during the year, the amount applied, appropriated, or expended and the purposes and objects for which such expenditures have been made, will be found in the report of the Finance Committee, duly certified by the Audit Company of New York.

The Council has received and considered a report giving the names and residences of those who have been admitted to membership in the Society since the last annual report, up to and including September 30th, and submits an abstract from this full report to be entered in the minutes of the proceedings of the annual meeting.

The Council has held six meetings during the year. For special duties the following appointments under the provision of the By-Laws and to meet special needs have been made:

On the John Fritz Medal Committee—Mr. John E. Sweet to serve for five years. Representatives of the Society on the Committee on the Engineering Building—Mr. James M. Dodge, Charles Wallace Hunt and F. R. Hutton. Messrs. Ambrose Swasey, F. R. Towne and H. H. Suplee, a conference committee to sit with appointees of the two other engineering societies who are to take part in building up the library in the Engineering Building. Messrs. C. H. Benjamin and R. H. Fernald to represent the Society as Honorary Vice-Presidents at the installation of President James of the University of Illinois.

Mr. Charles H. Haswell, the first incumbent of the office of Engineer in Chief of the United States Navy, now in his ninety-sixth year, was elected Honorary Member of the Society.

From Mr. E. R. Archer, of Tredegar Iron Works of Richmond, Va., was received the following letter with its accompanying gift:

"It has occurred to me that in connection with the life and work done for the Engineering Profession by that eminent engineer, John Ericsson, whose portrait hangs on the wall of the Society Hall, our members would appreciate having a contribution to our historical collection, as a connecting link with his life, the piece of armor plate I send by express to-day.

The plate was cut from the turret of the Monitor *Montauk*, one of the monitors built from the designs of John Ericsson during the Civil War in 1862. You will observe that the indentation made by the shot of the Confederate gun plainly shows that at that time there were no guns sufficiently powerful to penetrate this armor plate, but with the modern guns of to-day the shot would have passed through as easily as if made of tissue paper.

It is interesting to know that this monitor was struck ninety-five (95) times in the various actions in which she was engaged without serious damage during the Civil War, at Charleston and elsewhere.

I have recalled a part of the past history of this invulnerable giant in war, and now it may be proper to state that after many years of battle we who fought and tried to destroy are virtually converting this monitor into 'plough shares and pruning hooks' and other useful materials for the benefit of mankind.

With compliments of the season, I remain,

Yours very truly,

(Signed) E. B. ARCHER."

From Mr. Charles H. Oberge has been received, through Mr. H. H. Suplee, an example of the Nystrom Calculating Machine for the Society's historical collection.

From the Institution of Mechanical Engineers of Great Britain has been received a handsome illuminated address expressing from that body their thanks and recognition for the courtesies enjoyed at the hands of the Society during the successful joint meeting of the two societies in 1904.

Suitable letters of thanks have been transmitted in every case to the donors.

The Society has been asked to co-operate in the commemoration of the One Hundredth Anniversary of the first successful trip of Fulton's *Clermont* on the waters surrounding New York, and also in the commemoration of the Two Hundredth Anniversary of the birth of Benjamin Franklin.

Messrs. George W. Melville and Chas. H. Loring have been appointed to sit in conference in relation to the Fulton celebration,

and Mr. H. H. Suplee in connection with the Franklin Institute celebration.

The Committee consisting of Messrs. R. H. Fernald, M. L. Holman and W. H. Bryan, intrusted with providing and operating a headquarters for the Society in the Machinery Building of the St. Louis Purchase Exposition, reported the consummation of their duties with the close of the exposition and their opinion that while perhaps the maintenance of such headquarters at former expositions might have been worth while, it was their opinion from the conditions at St. Louis that it would not be advisable to repeat the experiment at subsequent expositions. The expense of this undertaking is reported in the financial statement.

The Council has approved the By-Laws submitted to it for its comment, from the Trustees of United Engineering Society, which is the corporation created under the laws of the State of New York to hold and administer the building, the gift of Mr. Carnegie for the uses of the engineering societies and the profession. The Council has also approved the form of agreement between the Trustees of this holding corporation and the American Society of Mechanical Engineers, and directed its President and Secretary to execute the necessary papers required by this arrangement and affix the seal of the Society to these documents.

The Trustees executed in January the mortgage for the real estate, covering $541,000, and in July the contract for the construction of the building whose plans had been under advisement for two years. At the date of the preparation of this report the foundation piers are in an advanced state.

The Council has had under advisement, on behalf of the Library Association, the question of the sale of the Society house at No. 12 West Thirty-first Street. Acting under competent advice from real estate experts offers of $115,000 have been refused, and the price set upon the property at this time is $125,000.

The Council has given careful consideration to the provision of Article C52 of its Constitution, giving authority to organize sections or groups. After full discussion of the questions brought up by this authorization it approved the following rules for the organization and conduct of such sections:

1. A Section of the Society shall consist exclusively of Honorary Members, Members, Associates and Juniors of the American Society of Mechanical Engineers.

2. A Section of the American Society of Mechanical Engineers may, with the approval of the Council, be organized in any city having not less than twenty-

five Members, Associates and Juniors resident within a radius of fifty miles. After the name of the Section adopted by the members may be added the words, "Section of the American Society of Mechanical Engineers."

3. Each Section shall be regularly organized with a Board of Trustees to be annually elected by the membership of the Section. The Board of Trustees shall annually elect a President, a Secretary, and a Treasurer from the members of the Board.

4. The management and financial responsibility shall rest entirely with the Section. The American Society of Mechanical Engineers assumes no responsibility for any act of the Section, but will require evidence that the Section will be self-supporting. The Section shall hold not less than six professional meetings during each year. Papers brought before the Section must be in writing or stenographically reported.

5. The Secretary of the Section shall promptly furnish to the American Society of Mechanical Engineers a copy of all papers presented to the Section, together with the discussion thereon. The American Society of Mechanical Engineers may present and read at its professional meetings or publish in its *Transactions* any paper or discussion it may select.

6. The Section shall not furnish papers or discussions to the public or technical press, until the American Society of Mechanical Engineers shall have first decided not to have it presented at its professional meeting or published in its *Transactions*.

7. The Council of the American Society of Mechanical Engineers may, at sixty days' notice, suspend or disband any Section. The title "Section of the American Society of Mechanical Engineers" shall not be used by any Society or persons without the consent of the American Society of Mechanical Engineers.

8. The American Society of Mechanical Engineers will furnish to its Sections copies of its professional papers to be read and discussed at the stated meetings of the Sections.

9. The American Society of Mechanical Engineers will print for the Section, at cost, such papers and discussions as the latter may order.

10. The Section shall not approve or adopt any commercial enterprise, device or standard.

On recommendation of the Finance Committee the Council has approved the appointment of a Cashier in the office of the Society who shall be intrusted with the duty under direction of the Secretary and the Finance Committee, of making the disbursements approved by the Finance Committee on the order of its Chairman. This cashier has been placed under $5,000 bond, and the Treasurer, instead of drawing every individual check as under the old system, draws on the order of the Finance Committee a single check covering the total of disbursements to be made to the Cashier's order, who thereupon makes the individual payments to the creditors of the Society.

On the recommendation of the Committee on Membership a form of application blank has been approved by the Council calling for a more detailed statement of professional achievements from

candidates, and the work actually accomplished in the positions which such candidates have held. This new blank will be distributed to all members with the Year Book for 1906.

The Council has made the Society a member of the National Fire Protection Association, which gives to the individual members of the Society a right to attend the conventions of this body for the discussion of matters relating to fire protection.

Messrs. Ira H. Woolson and John R. Freeman have been delegated to represent the Society officially at these meetings.

By vote of the Council the city of Chattanooga, Tenn., has been selected as the place for the spring meeting of 1906.

Under the direction of the Council and with the special co-operation of the President of the Society four reunions of members of the Society, resident within convenient distances of New York City, were held on the last Tuesdays of the months of January, February, March and April. The speakers on these evenings were Messrs. Rear-Admiral George W. Melville, "Epochs in Marine Engineering"; Worcester R. Warner and Prof. Wm. H. Burr, "The Conditions at Panama and the Reasons for a Sea Level Canal"; Dr. Howard T. Barnes, "Formation of Anchor Ice and Precise Temperature Measurements"; Gus C. Henning, "Diamonds and Diamond Tools."

Under the provision of Article 31 of the Constitution, the Council presents the following list of those elected to membership during the fiscal year 1904-5:

MEMBERS

Abbott, Chas. Carroll, Pittsfield, Mass.
Abbott, Fred'k Bancroft, Emporia, Kansas.
Ames, Geo. Edgar, Lowell, Mass.
Austin, William F., Long Island City, N. Y.
Bailey, Fred Wesley C., Cincinnati, Ohio.
Baker, Dickerson Gregory, Ilion, N. Y.
Baldwin, C. Kemble, New York, N. Y.
Barber, Clarence M., Detroit, Mich.
Baush, William H., Springfield, Mass.
Bayley, Wm., Springfield, O.
Bennett, George Lewis, Philadelphia, Pa.
Bentley, Ernest W., Pittsburg, Pa.
Bigelow, Charles H., Dallas, Texas.
Bigelow, Myron Julius, Stanford, Conn.
Branch, Joseph Gerald, St. Louis, Mo.
Browning, Charles, Jr., Sacramento, Cal.
Burleigh, Chas. Bates, Boston, Mass.
Burleigh, Wm. Freeman, Newark, N. J.
Carlsson, Carl A. V., Franklin, Pa.
Carroll, Alexander Winchester, Chrome, N. J.
Catt, Geo. Wm., New York, N. Y.
Champion, Harry W., Philadelphia, Pa.
Chickering, Kenton, Oil City, Pa.
Clemens, Chas. Wm., Birdsboro, Pa.
Coleman, Edgar Park, Buffalo, N. Y.
Crabtree, Frederick Herbert, Anaconda, Mont.
Crocker, Allen Swift, Rochester, N. Y.
Cummings, Wm. Warren, Woburn, Mass.
Davidson, Chas. Jackson, Milwaukee, Wis.
Dean, Edmund Willard, Dover, N. H.

Deuel, Harry Austin,
Pueblo, Colo.
Diefendorf, Willis H.,
Syracuse, N. Y.
Diman, Geo. Henry,
Lawrence, Mass.
Eckart, W. R., Jr.,
Palo Alto, Cal.
Elliott, Thomas,
Cincinnati, O.
Estrada, Rafael,
Guayabal, Cuba.
Ferris, Charles Edward,
Knoxville, Tenn.
Fischer, Adalbert,
Philadelphia, Pa.
Fowler, Fred Newton,
Springfield, Mass.
Fox, John Herbert,
Cleveland, O.
Franks, Fred'k Benjamin,
Bath, Pa.
Fry, Lawford Howard,
Philadelphia, Pa.
Fry, Thomas W.,
Claremont, N. H.
Gardner, Horace C.,
Chicago, Ill.
Gebhardt, George F.,
Chicago, Ill.
Gooding, Chas. S.,
Boston, Mass.
Gowie, William,
Williamson School P. O., Delaware Co., Pa.
Green, Morris M.,
Detroit, Mich.
Hale, Herbert Carlton,
Mineral Ridge, O.
Hall, Frederick Bellows,
New York, N. Y.
Harriman, Norman Follett,
Omaha, Neb.
Hart, Frederick,
Poughkeepsie, N. Y.
Hedemann, Christian J.,
Honolulu, Hawaii.
Hequembourg, Charles Guy,
Natchez, Miss.
Horton, S. Ellsworth,
Windsor Locks, Conn.
Hubert, Herman,
Liege, Belgium.
Hussy, Wm. Edgerly,
New York, N. Y.
Jackson, Charles J.,
Plainfield, N. J.
Jacobs, Henry William,
Topeka, Kansas.
Johnson, Paul Franklin,
Milwaukee, Wis.
Just, George A.,
New York, N. Y.
Kahn, Julius,
Detroit, Mich.
Kenerson, Wm. Herbert,
Providence, R. I.
Klinck, John Henry,
Pittsburg, Pa.
Kunhardt, Lewis Henry,
Boston, Mass.
Leach, Wm. Henry, Jr.,
Bridgeport, Ct.
Leland, Wm. Emmons,
San Francisco, Cal.
Long, Jeremiah Charles,
Boston, Mass.
McArthur, Alonzo Wm.,
Chicago, Ill.
McCrickett, Thomas F.,
Detroit, Mich.
Maltby, Geo. Beecker,
Cleveland, O.
Marshall, Ernest Woods,
New York, N. Y.
Martin, Fred'k Sheldon,
Brooklyn, N. Y.
Matlack, Ellwood V.,
St. Louis, Mo.
Maxfield, Howard H.,
Trenton, N. J.
Mohun, John L.,
Trenton, N. J.
Montague, Chas. Dwight,
Cold Springs, Putnam Co., N. Y.
Moore, Fred'k Clouston,
Auburn, N. Y.
Moyer, H. C.,
Coatesville, Pa.
Naylor, Charles William,
Chicago, Ill.
Nettleton, William Alpheus,
St. Louis, Mo.
Newton, Charles C.,
Philadelphia, Pa.
Norris, Wm. H.,
Philadelphia, Pa.
Nunn, Paul N.,
Niagara Falls, N. Y.
Ord, Henry C.,
Milwaukee, Wis.
Parker, John C.,
Philadelphia, Pa.
Perrigo, Oscar Eugene,
Boston, Mass.
Perrine, Frederic A. C.,
New York, N. Y.
Pingree, Edwin Daniel,
Providence, R. I.
Prince, Walter F.,
Harrison, N. J.
Rapley, Freder'k Harvey
New York, N. Y.
Rathbun, Geo. Jay,
Toledo, O.
Raymond, Alfred,
New Orleans, La.
Richart, Fred W.,
Carterville, Ill.
Roberts, Edmund W.,
Clyde, O.
Robinson, James R.,
Monongahela, Pa.
Roper, Norman Brownell,
Cananea, Sonora, Mexico.
Ruggles, Wm. Barker,
New York, N. Y.
Rust, William F.,
Joliet, Ill.
Schlesinger, George,
Charlottenburg, Berlin, Germany.
Sherrerd, John Maxwell,
High Bridge, N. J.
Shipman, Robert Lee,
Ithaca, N. Y.
Starr, John Edwin,
New York, N. Y.
Stebbins, Albert C.,
Plainfield, N. J.
Tait, Roderick H.,
St. Louis, Mo.
Thullen, L. H.,
Edgewood Park, Pa.
Town, Frederic Edw.,
New York, N. Y.
Trinks, Chas. Leopold W.,
Pittsburg, Pa.
Tucker, Frank Stevenson,
E. Newark, N. J.
Wait, Henry Heileman,
Chicago, Ill.
Wallace, Ross S.,
Peoria, Ill.
Wells, George Augustus, Jr.,
New York, N. Y.
Whiton, Lucius Erskine,
New London, Ct.
Williston, Belvin Thomas,
Boston, Mass.
Wilmerding, Chas. Henry,
Chicago, Ill.
Wilson, Birton N.,
Fayetteville, Ark.
Zimmerman, Oliver B.,
Milwaukee, Wis.

ASSOCIATES.

Berry, Edgar Henry, Ilion, N. Y.
Breckenridge, C. E., San Francisco, Cal.
Bridge, James Weldon, Connellsville, Pa.
Bronaugh, Will Logan, Chicago, Ill.
Caracristi, Virginius Z., Granite, Va.
Carrier, Willis Haviland, Buffalo, N. Y.
Chisholm, John James, New York, N. Y.
Clark, Edward Lord, Boston, Mass.
Einstein, Alfred Charles, St. Louis, Mo.
Flinn, Charles Forrest, Chicago, Ill.
Gately, Philip J., New York, N. Y.
Gray, John Lamont, Williamstown, Victoria, Australia.
Hill, Robt. J., Chicago, Ill.
Hogan, Patrick Henry, Boston, Mass.
Hume, William H., Chattanooga, Tenn.
Jahncke, Ernest Lee, New Orleans, La.
Jurgensen, Jess Christian. New York, N. Y.
Kreutzberg, Otto August, Chicago, Ill.
Little, Chas. Henry, Cleveland, O.
McArthur, Harold, Cleveland, O.
MacArthur, Robert, Jr., New Haven, Conn.
Morgan, Lewis Henry, Ridgway, Pa.
Ostrander, Allen Edward, New York, N. Y.
Polk, Wm. Anderson, New York, N. Y.
Price, Melvin, Lincoln, Neb.
Prince, John Walter, Harrison, N. J.
Rawson, Louis W., Worcester, Mass.
Ray, David H., New York, N. Y.
Rutherford, Eugene W., Brooklyn, N. Y.
Sanders, Lewis. West Lynn, Mass.
Schneller, Geo. Otto, Ansonia, Conn.
Schumaker, John Staubley, Ansonia, Ct.
Serrell, Harold, New York, N. Y.
Smith, Geo. Marshall, Pittsburg, Pa.
Stehlin, Joseph, New York, N. Y.
Stevens, Jesse F., Boston, Mass.
Traver, Wilber H., Chicago, Ill.
Vandemoer, John, Denver, Colo.
Von Ammon, Siegfried, London, W. C., England.
Wilder, Stuart, Ossining, N. Y.
Wile, Julius I., Rochester, N. Y.
Willhöfft, Friedrich Otto, Potsdam, N. Y.
Wilson, Victor Tyson, Urbana, Ill.

JUNIORS.

Allen, F. Ramsey, New York, N. Y.
Allen, Walter Cleveland, Stamford, Conn.
Alsberg, Julius, New York, N. Y.
Ancona, John F., Proctor, Vt.
Austin, Adolph Odell, New York, N. Y.
Bacon, Charles James, New York, N. Y.
Barnes, Chas. Ballou, Milwaukee, Wis.
Barstow, Francis Loring, Mittineague, Mass.
Baylis, Arthur Raymond, New York, N. Y.
Beach, Harold Kenney, Ansonia, Conn.
Been, Peter H., Milwaukee, Wis.
Bibbins, James Rowland, E. Pittsburg, Pa.
Bornholt, Oscar, Chicago, Ill.
Bradshaw, Grant D., Joliet, Ill.
Brooks, Paul Raymond, New York, N. Y.
Buckler, Albert Hammond, East Pittsburg, Pa.
Church, Herbert B., Bloomfield, N. J.
Cole, Arthur W., Orono, Me.
Comly, G. Norwood, Syracuse, N. Y.
Cook, Thos. Fowke, New York, N. Y.
Cooke, Saint Geo. Henry, Philadelphia, Pa.
Corp, Charles I., Lawrence, Kansas.
Cox, Frank Gardner, London, E. C., England.
Cushman, Arthur Wesley, Brightwood, Mass.
Dawley, Clarence A., Easton, Pa.
Dixon, Horace H., Chicago, Ill.
Douglas, Courtney Carlos, Schenectady, N. Y.
Earle, Samuel Broadus, Clemson College, So. Ca.
Eberhardt, Elmer Gould, Newark, N. J.
Enslen, Eugene Flynn, Jr., New York, N. Y.
Fleming, Wills Maine, Holyoke, Mass.
Garza-Aldape, J. M., Torreon, Coah., Mex.
Green, Charles Henry, St. Louis, Mo.
Griffiths, Leonard L., Brooklyn, N. Y.
Hamilton, Edward Waterman, Oakville, Ct.
Hamilton, Thomas Smith, Louisville, Ky.
Haney, James Briggs, Washington, D. C.

Harkins, Robert R., Pittsburg, Pa.
Harlan, Orla K., Schenectady, N. Y.
Harrison, Edwin Sterne, New York, N. Y.
Hawks, Arthur Stearns, E. Pittsburg, Pa.
Helvey, Clarence Harman, Chicago, Ill.
Helvey, Geo. Stanley, Hamilton, O.
Hirshfeld, Clarence Floyd, Ithaca, N. Y.
Holmes, Arthur, Syracuse, N. Y.
Holmes, Andrew A., Reading, Pa.
Horne, Convers Francis, New York, N. Y.
Hunting, Eugene Nathan, Pittsburg, Pa.
Hurley, Daniel, Harrison, N. J.
Hutchins, Harry Crocker, Brooklyn, N. Y.
Jackson, Roscoe B., Lansing, Mich.
Jones, Daniel Lanning, Ampere, N. J.
Keith, Robert R., Milwaukee, Wis.
Keith, Thomas, New York, N. Y.
Kent, Robert Thurston, Cleveland, Ohio.
Kingsbury, Ralph E., Cincinnati, O.
Kneip, Walter Francis, Ypsilanti, Mich.
Lang, Charles, New York, N. Y.
Lathrop, William Frederic, Milwaukee, Wis.
Leutwiler, Oscar A., Champaign, Ill.
Lockwood, Rutherford T., Bayonne, N. J.
Lucas, Henry Van Noye, Jr., Huntington, W. Va.
Lyman, Elihu Root, Chicago, Ill.
McBain, Wm. Coryell, Youngstown, Ohio.
McGregor, Alex. Grant, Anaconda, Mont.
McMeans, O. E., Indianapolis, Ind.
Marsh, Thomas Alfred, Mansfield, Ohio.
Meaker, Guy Larned, Chicago, Ill.
Morrison, James, Cincinnati, O.
Murphy, Benj. Stewart, Altoona, Pa.
Murphy, Edw. Thos., New York, N. Y.
Murrie, John Lester, New York, N. Y.
Myers, Curtis Clark, Syracuse, N. Y.
Newell, Charles Z., New York, N. Y.
Northrup, Francis B., New York, N. Y.
Northrup, Lewis Mulford, Hudson, N. Y.
Owen, Ira June, Chicago, Ill.
Palmer, Virgil Maro, Hagerstown, Maryland.
Pedersen, Henrik Greger, Milwaukee, Wis.
Phetteplace, Thurston M., Providence, R. I.
Proctor, Redfield, Jr., Proctor, Vt.
Rauch, John Dodds, Logansport, Ind.
Schneider, Paul E., New York, N. Y.
Sherburne, Kenneth, West Lynn, Mass.
Shiebler, Marvin, New York, N. Y.
Smith, Ellis Burton, Jersey City, N. J.
Smith, Roy Brooke, Columbus, Ohio.
Stratton, Harry Frost, Tiffin, Ohio.
Thurn, Theodore, Yokohama, Japan.
Titcomb, Roland Elbert, Johnstown, Pa.
Trautschold, Reginald, Montclair, N. J.
Valentine, Warren P., Plainfield, N. J.
Weeks, Paul, Philadelphia, Pa.
Weymouth, Clarence R., San Francisco, Cal.
Wheeler, Seth, Jr., Wyandotte, Mich.
Whipple, Wm., Cinclare, La.
Winger, Stanley D., Columbus, O.
Wolff, Herbert Wm., St. Louis, Mo.
Wyer, Samuel S., Columbus, O.

Losses by death from the membership during the current year have been as follows:

Chas. L. Bailey, James T. Boyd, Theo. F. Burgdorff, R. G. Collins, Geo. E. Dixon, Carl F. Eicks, Andrew Fletcher, G. W. Frank, G. A. Gray, R. C. Greer, Wm. A. Heywood, D. T. Matlack, T. R. Morgan, Wm. O. Mundy, E. H. Parks, Geo. H. Perkins, Francis Reuleaux, William Sellers, Fred. W. Taylor, Jr., Wm. G. Vernon, C. H. Wellman, C. M. Wilkes, H. F. Witte, Horace W. Wyman.

The council would call particular attention to the reports of its standing committees which appear as appendices herewith.

APPENDIX 1.

REPORT OF THE FINANCE COMMITTEE.

The Finance Committee presents the following report to the Council.

The Committee submits the financial statements hereto appended, which have been prepared by the accountant of the Society, and submitted to the Audit Company of New York for scrutiny and audit. Their report is appended. The Financial Statements include:

Sheet A, a Statement of Cash Account, Receipts and Disbursements.

Sheet B, a Statement of Income and Expenditure.

Sheet C, a Balance Sheet of Assets and Liabilities.

Sheet D, a Tabular Statement of Changes in these Assets and Liabilities as compared with the condition September 30, 1904.

The Committee submits also a Sheet of estimated receipts and expenditures for the coming year, 1905–6, based on the assumption that no important changes are made in the direction or extent of expenditure for the coming year as compared with the year which has closed. This estimate is designated as Sheet E.

The table below will show clearly the net gain for the fiscal year 1904–5 which has just closed with respect to the Income and Expense Accounts of the two years. At September 30, 1904, the Income and Expense Accounts showed that the expenses had exceeded the income for the year 1903–4 by $2,005.17, whereas for the year 1904–5 the income has exceeded the expenses by $1,450.61, making the net gain on income and expense for the year ending September 30, 1905, the sum of these two amounts or $3,455.78, which represents the total gain for the year. As clearly shown in the table below, the net gain in income for the year 1904–5 was $1,094.43, while the net gain due to decreased expense was $2,361.35, the sum of which two amounts equals the total net gain for the year 1904–5, *i.e.* $3,455.78. In connection with the gains through decrease in expenses the principal gain on the House Account has been in the matter of Repairs and Renewals which cost, for 1903–4, $1,561.30, while for the year just ended, 1904–5, their cost was but $625.53. In connection with the net gain through decrease of expense for meetings, the principal gain is in the cost of the Spring Meeting, as the Spring Meeting of 1904 cost $1,305.39, whereas the Spring Meeting of the year 1904–5 cost $597, the decreased expense being due to the fact that the Spring Meeting for the year 1904–5 was held nearer New York and was not as largely attended as the one for 1903–4, which latter meeting was the Joint Meeting of the Institution of Mechanical Engineers of Great Britain and this Society, the expense in connection with which was unusually heavy.

Last year the inventory value of the stock of *Transactions* showed a decrease of $614.02, whereas this year the inventory of September 30, 1905, showed an increase of $1,112.68. This is due to the fact that a new edition of Volume 24 was issued this year which increased the value of the stock of *Transactions* on hand at the end of the year. Under ordinary conditions the inventory at the end of a year will naturally show a loss over that at the end of the previous year.

TABLE SHOWING NET GAIN, 1904-5.

Income and Expense.

Income.	1903-4	1904-5	Gain
Gross Income	$41,520 73	$42,615 16	$1,094 43
Gross Expense	43,525 90	41,164 55	2,361 35
Total Gain 1904-5 over 1903-4			$3,455 78
That is—			
Excess Expense over Income	2,005 17		2,005 17
Excess Income over Expense		1,450 61	1,450 61
Total Gain 1904-5 over 1903-4			$3,455 78

The make up of this Gain is as follows:

	Income Increase	Income Decrease	Net Gain
Dues Income, 1904-5	$1,993 01		
Initiation Fee, 1904-5	17 00		
Other Sources:			
Decrease in Sales and Rentals		$2,069 44	
Less Increase Stock *Transactions* on hand, due to new edition of Volume XXIV at cost price		1,112 68	
		$956 76	
Less Increase in Miscellaneous Income		41 18	
	$2,010 01	$915 58	—$1,094 43

Expense.	Expenditure Increase	Expenditure Decrease	
Publications, including new edition Volume XXIV		$148 34	
House, Total Expense		904 28	
Meetings, Annual, Spring and Monthly		821 96	
Miscellaneous, Certificates, Cards, Expert Fees, etc., etc		274 60	
Office Expenses, Salaries, Year Books, and Catalogue and Circulars	$175 05		
Library, Total Expense	226 80		
Stock *Transactions*, Increase in Inventory value due to new edition Vol. XXIV		614 02	
	$401 85	$2,763 20	—$2,361 35
Total Gain 1904-5 over 1903-4 as shown above			$3,455 78

The Committee would call especial attention to the itemized Statement of the Library Development Fund, the Reserve Fund, the Trust Funds, and the George W. Weeks Legacy Fund as shown under Liabilities in the Combined Balance Sheet (Sheet C), and also to the footnotes at the bottom of the Tabular Statement (Sheet D), and in the remarks on same which follow that sheet which refer to the accumulation which has been made to these funds during the three last years, from which it will be seen that a total asset of $7,-009.48 has been created by cash deposited in Savings Banks, offsetting the total liability of the same amount to these funds, in addition to which there has been a cash investment from the Reserve Fund, Initiation Fees, of $7,971.11 in the new Engineering Building. This showing has been made possible largely by the great number of new members elected and the initiation fees and dues received from them.

The Committee would submit also computations which have been deduced from the accounts of the current fiscal year, as follows:

(1) Total members as per July, 1905, catalogue	2,915		
Add new members who have paid since then	9	—2,924	
Deduct for members who have paid no dues; Life Members	109		
Honorary Members	20		
Deaths and resignations, without payment	8		
Lapsed memberships	58		
Members who have not paid current year at September 30, 1905	202	— 397	
Paying membership, 1904–5		2,527	
(2) Total income exclusive of 1 per cent. from dues carried to Library Development Fund, 90 per cent. from initiation fees, entire life membership receipts, carried to Reserve Fund, and entire Sinking and Fellowship Fund, subscriptions to Mechanical Engineers' Library Association		$42,615 16	
Income per paying member (computed)		16 86	
Income per paying member, from dues only (computed)		14 88	
(3) Total expense incurred year October 1, 1904, to September 30, 1905, less cost operating house ($3,768.27), mortgage interest ($1,402.50), repairs and renewals ($625.53), depreciations house and furniture ($427.27)—$34,940.48:			
(4) Total expense incurred for publications, October 1, 1904, to September 30, 1905		14,770 88	
(5) Total expense incurred for salaries in Society's office same period		10,200 00	
(6) Total expense incurred for all other accounts except house		9,969 60	—$34,940 48

Brought forward			$34,940 48
(7) Total expense incurred for house, including interest on mortgage, repairs and renewals and depreciations		6,224 07	
Deduct income earned from rent of rooms and hall		870 50—	5,353 57
Net expense incurred for year 1904–1905			$40,294 05
(Gross expense, $41,164.55 less rental income $870.50, equals $40,294.05.)			

Expenses incurred per paying member, October 1, 1904, to September 30, 1905:

(8) For all purposes including house		$15 94
(9) For house operation including interest and repairs and renewals and depreciations		2 11
(10) For all purposes exclusive of house		$13 83
(11) For publications, printers' work, engraving, binding and distribution	$5 85	
(12) For salaries in Society's office	4 03	
(13) For all other expenses except house	3 95—	13 83
(14) For house operations exclusive of mortgage interest, repairs and renewals and depreciations		1 14
(15) For house operation exclusive of mortgage interest, but including repairs, renewals and depreciations		2 11
(16) For operating library		46
(17) For Postage, circulars, catalogues, and stationery and printing in Society's office		2 69
(18) For meetings, and all other expenses not otherwise allotted above		78

Comparative income earned with expense incurred per paying member:

*Income earned from all sources per paying member, per (2)	$16 86
Income earned from dues only per paying member, per (2)	14 88
(19) Excess income earned from all sources, per paying member over expense incurred all purposes, per paying member	92
(20) Excess expense incurred all purposes, per paying member over income earned from dues alone, per paying member	1 06

From the above it would appear that the Income from dues per member, $14.88, is less than the expenditure per member, $15.94.

The phenomenal growth of the Society and the unusual sums received from

* Ninety-nine per cent. of dues and ten per cent. of Initiation Fee receipts considered as income. Receipts for Life Memberships and subscriptions to Fellowship and Sinking Funds not considered as income.

initiation fees for the past two years should be noted in connection with the sheets of accounts presented.

The net expenses which by Resolution of the Council this Committee audits include the salaries of the Secretary and of the office staff, and the expense for miscellaneous accounts passing through the administrative office of the Society. The elements of these two channels of expense have been as follows:

Total Appropriation for work of Finance Committee for the fiscal year 1904–5	$17,755 50
Total Net Expense for same year	17,349 00
Decrease Net Expense from Appropriation	$406 50

Detail of Expense.

Salaries:

Secretary	$3,600 00
Cashier, Assistant to Treasurer and Accountant	2,500 00
Assistant to Secretary	2,000 00
Stenographer	840 00
Stenographer	540 00
Mail Clerk	720 00
	$10,200 00

Detail of Expense other than Salary.—*Finance Committee.*

Accounts.	Net Expense. Detail.	Totals.
Certificates and Introduction Cards:		
Certificates, including distribution	$234 79	
Cards, including distribution	18 84	$253 63
Distribution of badges	39 33	39 33
Circulars, printing and distribution:		
General	371 51	371 51
Catalogues:		
Composition, press work and paper and binding	2,630 34	
Postage and distribution	347 41	2,977 75
Office Account:		
Stationery and printing	564 67	
Postage, General	641 02	
Telephone, telegraph and messengers	89 13	
Supplies	319 04	
Incidentals	231 05	1,844 91
Interest on mortgage	1,402 50	1,402 50
Legal Expenses and Expert Fees:		
Legal Expenses	00 00	
Expert Fees	105 00	105 00
Headquarters, St. Louis Exposition	134 37	134 37
Committee Work	20 00	20 00
Total		$7,149 00

The assessment of this Society for the new Engineering Building for the fiscal year 1904–5 was $8,000, but the Trustees of the United Engineering Society, the holding Society, have approved of the payment to each Founder Society of interest at four per cent. on the moneys it advances on account of this $8,000 prior to July first, when it was due; in other words, for an equalization of partial payments this Society received a credit by such interest of $28.89, making the net cost to the Society for its assessment for the year 1904–5 $7,971.11, the cash for the payment of this assessment having come from the Reserve Fund, as follows:

Life Membership	$1,311 84	
Initiation Fees	6,659 27	—$7,971 11

The Committee would also submit the following Statement which shows clearly what may be called the expected income from dues, that is, the income which the Society would have derived from dues if all men who were members during the year 1904–5, and those elected during that year had paid their dues before September 30, 1905, and the amount actually collected of same, the amount written off of same and the reasons why it was written off, and the balance considered as a good asset of the arrears of dues for the year 1904–5, *i.e.* $2,800.81; as well as the amount owed us by men elected during the year 1904–5, but who have not as yet paid either their initiation fees or dues, *i.e.* $65.

Statement showing Expected Income from Dues Fiscal Year 1904–5.

Amount that should have been earned from arrears of dues for 1903–4, outstanding and considered good at October 1, 1904	$2,882 48	
Amount that should have been earned from current dues 1904–5, if all members at October 1, 1904, all men elected members during the year 1904–5 had paid their dues, and including all men who returned to active membership from Suspended List in same year	40,195 00	
Total Expected Income from dues	$43,077 48	—$43,077 48

Actual Income from dues, fiscal year 1904–5:	
Arrears of dues (1903–4)	$2,052 48
Current dues (1904–5)	36,024 19
	$38,076 67
Arrears of dues 1904–5 at September 30, 1905, considered as a good asset	2,800 81
Total	$40,877 48

Brought forward			$40,877 48	
Dues written off:				
Arrears at September 30, 1904, time limit of payment exceeded	$760 00			
Arrears at September 30, 1904, deaths and resignations	70 00	$830 00		
Current 1904-5 at September 30, 1905, as time limit of payment exceeded	935 00			
Current ditto, account, deaths and resignations	320 00			
Current ditto, account, member made a Life Member by Council without fee	15 00	1,270 00		
Current ditto, members elected who did not pay up in six months		35 00		
Total written off			2,135 00	
			$43,012 48	
To Balance:				
Dues unpaid by members elected on Spring Ballot 1905, at September 30, 1905			65 00	
			$43,077 48	$43,077 48

A scrutiny of Sheet D, Tabular Statement of Changes in Assets and Liabilities will show that the surplus at the end of the year 1904-5 is $9,400.44 greater than the surplus at the end of the fiscal year 1903-4; that is, that our assets at the end of this year are $9,400.44 greater than they were at the end of last year, and the causes for this increase are explained in detail in the Remarks in Comment on the Tabular Statement of Changes in Assets and Liabilities which follows Sheet D.

Respectfully submitted,

E. D. MEIER,
DAVID TOWNSEND,
MILTON P. HIGGINS,
ANSON W. BURCHARD,
ARTHUR M. WAITT,
} *Finance Committee.*

APPENDIX 2.

REPORT OF THE COMMITTEE ON MEETINGS.

Below will be found a Statement showing the total appropriation by the Council for the work of this Committee and the total net expense for the year, which shows that the Committee has met the expenses of the meetings for the sum of $958.61 less than the appropriation allotted therefor by the Council.

Total Appropriation for work of Meetings Committee for the fiscal year 1904–5	$5,900 00
Total Net Expense for same year	4,941 39
Decrease Net Expense from Appropriation	$958 68

DETAIL OF EXPENSE.—*Meetings Committee.*

Accounts.	Net Expense. Detail.	Total.
Meetings.		
Monthly	$244 33	
Annual	572 07	
Spring	597 00	
Advance Papers	2,650 27	
Stenographers' Fees	242 95	
Programs, Registers, etc.	634 70	
Total		$4,941 32

Respectfully submitted,

ARTHUR L. WILLISTON,
W. S. ACKERMAN,
WALTER M. MCFARLAND,
CHARLES WHITING BAKER,
CALVIN W. RICE,
Meetings Committee.

APPENDIX 3.

REPORT OF THE MEMBERSHIP COMMITTEE.

The Membership Committee would report that it has held frequent meetings during the year for the consideration of applications for membership and has received and considered 380 such applications of which it has passed to ballot 333 prior to September 30, 1905. The Statement below shows clearly not only the number of applications acted upon and passed to ballot but the Income which would have been derived from the men elected, provided all of them had paid up before September 30, 1905, and also what was actually received from them both for initiation fees and dues.

Total Applications acted on during 1904–5	380			
Applications Deferred or Rejected		47		
Total Applications passed to ballot 1904–5			333	
Total men elected, including promotions, 1904–5				332
Election Declined				1

	Initiation Fees.		Dues.	
Total amount due from men elected, fiscal year 1904–5	$6,200 00		$3,635 00	
Total Income—Payments from men elected, fiscal year 1904–5		$6,020 00		$3,530 00
Amount due us from men elected whose election was void on account of non-payment within time limit—dropped		75 00		40 00
Amount due us and unpaid at September 30, 1905, from men elected Spring ballot 1905—time limit not expired		105 00		65 00
	$6,200 00	$6,200 00	$3,635 00	$3,635 00

Actual Expense of election per member elected 1904–5	$2 96
Actual Income per member elected during the year of his election—Initiation Fee and Dues	28 76

Below is a Statement showing the total money appropriated by the Council for the work of this Committee, the total expense for the year, which it will be noted was $116.75 less than the total appropriation.

Total Appropriation for work of Membership Committee for fiscal year 1904–5	$1,100 00
Total Net Expense for Admission Circulars and Correspondence, including postage thereon for the fiscal year 1904–5	983 25
Decrease Net Expense from Appropriation	$116 75

Respectfully submitted,

Francis H. Stillman,
Wilfred Lewis,
Ira H. Woolson,
Jesse M. Smith,
Henry D. Hibbard,
Membership Committee.

APPENDIX 4.

REPORT OF THE COMMITTEE ON PUBLICATIONS.

The Publication Committee presents the following report of matters under its direction.

At the close of the last fiscal year it was estimated that the sum of $6,610 would be needed to complete Volume 25. The amount actually required to complete the Volume proved to be $227.28 more than the amount reserved, *i.e.* $6,837.28, making with the expense in the previous year the total cost of Volume 25—$15,146.50. The Volume contained 1,155 pages and the total cost per copy was $5.04, the total cost for Volume 24, the Volume for the previous year, having been $5.62 per copy of 1,563 pages.

The expense for Volume 26, the Volume for the year 1904–5 has been to date $7,440.88, and it is estimated to complete the Volume a sum will be required of approximately $6,590, in which event the total cost of the Volume will be about $14,030.

The items composing the expense incurred under direction of the Publica-

tion Committee are given below. This total, however, does not include the cost of advance papers and stenographers' fees, which expense is in the hands of the Committee on Meetings and which is shown in the detailed expense of that Committee in their report.

Total Appropriation for work of Publication Committee for fiscal year 1904-5	$14,175 00
Total Net Expense for same year	11,877 66
Decrease Net Expense from Appropriation	$2,297 34

DETAIL OF EXPENSE.—*Publication Committee.*

ACCOUNTS.	NET EXPENSE. Detail.	Total
Transactions:		
Revised Papers	$1,274 40	
Engraving	997 89	
Composition and electrotyping	4,334 30	
Binding	2,816 40	
Boxing plates	22 50	
Postage and express, distribution	1,430 45	
Storage including insurance	261 72	
* New edition Volume XXIV	740 00	
Total		$11,877 66

Respectfully submitted,

GEORGE M. BASFORD,
HENRY SOUTHER,
C, J. H. WOODBURY,
HENRY HARRISON SUPLEE,
WALTER B. SNOW,

Publication Committee.

APPENDIX 5.

REPORT OF THE HOUSE COMMITTEE.

This Committee which is entrusted with directing the expense for the maintenance of the house of the Society, which is its office headquarters also, and which provides for certain rooms in the upper floors at the service of visiting members for short stays in the city, would submit the following report.

Total Appropriation for work of House Committee for the fiscal year 1904-5	$4,800 00
Total Net Expense for same year	4,554 17
Decrease Net Expense from Appropriation	$245 83

* Made necessary by the issue of an unusually limited edition at the time when close economy was thought advisable. Made from the stereotype plates.

DETAIL OF EXPENSE.—*House Committee.*

ACCOUNTS.	NET EXPENSE. Detail.	Totals.
House Account:		
Gas and Electric Light	$725 69	
Fuel	367 70	
Janitor's Supplies	220 50	
Laundry	407 62	
Insurance	80 30	
Wages	1,773 00	
Incidentals	193 46—	$3,768 27
Repairs and renewals:		
House	249 85	
Furniture	375 68—	625 53
*Additions:		
House	....	
Furniture	160 37—	160 37
Total		$4,554 17

From the above report it will be seen that the Committee has met the expenses of the year in connection with the house at a sum $245.83 under the appropriation allowed them by the Council at the first of the year for their work.

The income from use of house by members and others has been $870.50. This is less than in previous years.

The Income and Expense Account for the year clearly shows the expense for the operation and maintenance of the house and the amount which has been received from room rent during the year, and it will be seen that the net cost of operating the house for the year exclusive of repairs and renewals, depreciations and mortgage interest was $2,897.77, whereas the cost inclusive of mortgage interest was $4,300.27, and the total expense including the above and repairs and renewals and depreciations has been $5,353.57.

The House Committee employs for the conduct of the house a janitor at $45, an assistant janitor at $40, and a maid at $20, while the house matron receives $40, for her services, in addition to which she receives other compensation for her services in the Library.

Respectfully submitted,

GEORGE J. FORAN,
RALPH L. MORGAN,
THOMAS R. ALMOND,
JOHN C. KAFER,
J. WALDO SMITH,
} *House Committee.*

* The Expenditure for Additions, both House and Furniture, passes through the hands of this Committee, but the net expense of same is treated as an increase of assets and not as an expense through Income and Expense Account.

APPENDIX 6.

REPORT OF LIBRARY COMMITTEE.

The Library Committee presents the following report. The Library has been open every day between the hours of 10 A.M. and 10 P.M., except Sundays, Thanksgiving, Christmas and New Year, and excepting the month of August, when it was closed for cleaning and overhauling.

An average of ten persons per day have made use of the Library during the busy season and about one-half this number during the vacation period of the summer months. This makes a total of between 2,500 and 3,000 visitors for the year.

The use of the Library during the evenings has been particularly noteworthy.

The number of volumes in the Library September 30, 1905, is as follows:

Books	9,144
Pamphlets	4,055
Maps	33

The book value of the Library at the same date is $12,588.67, which is the sum of the valuation at the end of the previous year's report, and of additions made during the year.

The additions to the Library in the form of exchanges which have been received as the equivalent of the annual volume of the Society's *Transactions*, have amounted to $488, there have been received through the exchange account with D. Van Nostrand books to the value of $160.75, while the Committee has expended for the purchase of books $55.40, making the total amount expended for additions to the Library for the year $704.15. There has also been an expense of $287.20 for binding periodicals and pamphlets received in exchange from other organizations or publishing agencies. There remains standing on the Society's books a charge against the publishing house of D. Van Nostrand Company for *Transactions* furnished them for which the Society is to purchase books from them as required, amounting to $224, and with the house of Spon & Chamberlain there is a similar balance in our favor of $16.50.

Below is given a detailed Statement of the expense of this Committee on behalf of the Library, and the appropriation made by the Council for the year, from which it will be seen that the Committee has expended $336 less than the original amount appropriated for its work.

Total Appropriation for work of Library Committee for the fiscal year 1904–5	$1,557 00
Total Net Expense for same year	1,221 00
Decrease Net Expense from Appropriation	$336 00

DETAIL OF EXPENSE.—*Library Committee.*

Accounts.	Net Expense. Detail.	Total.
Library:		
Book purchase	$55 40	
Expense	278 40	
Salary	600 00	
Binding	287 20	—$1,221 00

Respectfully submitted,

HENRY R. TOWNE,
E. A. UEHLING,
W. D. FORBES,
FREDK. M. WHYTE,
GEO. F. SWAIN,
} *Library Committee.*

APPENDIX 7.

AUDIT OF THE SOCIETY ACCOUNTS.

The Finance Committee would report that they have caused an audit of the books of the Society to be made for the fiscal year 1904–5 and selected the Audit Company of New York for this duty. The report of the Audit Company is as follows:

THE AUDIT COMPANY
OF NEW YORK.
43 Cedar Street.

COL. E. D. MEIER,
Chairman Finance Committee,
American Society of Mechanical Engineers,
12 West 31st Street, New York City.

Dear Sir:

Agreeably to your request, we have audited the books and accounts of the *American Society of Mechanical Engineers* for the fiscal year ended September 30, 1905. The results of this audit are presented, attached hereto, in the form of duly certified printed copies of the accounts prepared by Mr. Francis W. Hoadley, Cashier.

We found that the books and accounts of the Society had been kept in a thoroughly systematic manner. The detailed information contained therein is explicit and complete.

We certify that the Balance Sheet presented herewith is a correct exhibit of the position of your Assets and Liabilities on September 30, 1905, and that the accompanying Income Account is also a correct exhibit of your operations for the fiscal year ended September 30, 1905, as shown by said books and accounts.

Very truly yours,

THE AUDIT COMPANY OF NEW YORK.

E. T. PERINE,

General Manager.

NEW YORK, *October* 24, 1905.

SHEET A.

COMBINED STATEMENT OF CASH ACCOUNT.

FISCAL YEAR, 1904–1905.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

October 1, 1904, *to September* 30, 1905.

Receipts.

Date	Item		
1904. Oct. 1.	To Cash on hand		$84 66
1905 Sept. 30.	" Cash receipts other than Trust Fund Subscriptions during the year	$49,405 17	
"	" Cash receipts, Trust Fund Subscriptions of Mechanical Engineers' Library Association, Fellowship and Sinking Funds of said Association	370 00	
"	" Cash withdrawn from Union Square Savings Bank	4,000 00	
"	" Cash from United Engineering Society, interest rebate	28 89	
			53,804 06
			$53,888 72
1905. Oct. 1.	To Cash in East River National Bank, New York	$26 36	

Disbursements.

Date	Item		
1905. Sept. 30.	By Disbursements for Expenses of Fiscal Year 1903–4 to complete Volume 25 of the *Transactions* (see Sheet C, Surplus)	$6,837 28	
"	" Disbursements for Expenses of Fiscal Year 1904–5, all bills received to date paid	36,361 96	
			$43,199 24
"	" Payment to United Engineering Society, investment in land for United Engineering Building		8,000 00
	Total Disbursements		$51,199 24
"	" Money deposited in Savings Banks, 1904–5:		
	Trust Fund—M. E. L. A.	$370 00	
	Reserve Fund—A. S. M. E.	2,293 12	
			2,663 12
"	" Balance: Cash in East River National Bank, New York		26 36
			$53,888 72

Certified to as correct.

FRANCIS W. HOADLEY,
Cashier and Assistant to Secretary and Treasurer.

SHEET E.

ESTIMATE OF RECEIPTS AND EXPENSES, FISCAL YEAR 1905-1906.

October 1, 1905, *to September* 30, 1906.

Estimated Receipts.

Dues, 99 p. c. Gross Receipts		$39,000	
Initiation Fees, 10 p. c. Gross Receipts		500	
Sales Account—Publications		2,400	
House Account—Rent Sleeping Rooms	$800		
" " " Hall	55		
		855	
Misc. Sales of Electros, etc		25	
Estimated Receipts 1905-6		$42,780	
			$42,780

*Estimated Expenses.**

House Account		$3,850	
Interest on Mortgage		1,402	
Repairs and Renewals		500	
Additions—House and Furniture		300	
Certificates and Membership Cards		250	
Transactions—			
Volume 26, cost to complete	$6,590		
" 27, cost for work done during 1905-6	8,000		
		14,590	
Circulars		2,300	
Meetings		1,800	
Year Books and Pocket List of Members		3,000	
Salaries		10,980	
Office Account		1,900	
Repairs and Distribution of Badges		50	
Library Expense Account (including Binding and Librarian's Salary)		1,250	
Interest on Bank Loans		....	
Committee Work—Research		250	
Legal Expenses and Expert Fees—Legal Expenses		50	
" " " " " Expert Fees		150	
Estimated Expenses, 1905-6			$42,622
By Balance:			
Estimated Excess Receipts over Expenses			158
			$42,780

* These are gross expenses, representing cash payments which it is estimated will have to be met in the year.

REMARKS IN COMMENT ON TABULAR STATEMENT OF CHANGES IN ASSETS AND LIABILITIES

Prepared by FRANCIS W. HOADLEY, Cashier.

The tabular Statement on previous page shows the financial condition at the beginning and the end of the fiscal year 1904–5, the changes in each of the accounts due to the transactions of the year, and resulting changes of Assets and Liabilities. These may be summarized as follows:

Surplus, September 30, 1904 (as reported)		$79,802 73
Add amount of Reserve Fund cash invested in Engineering Building (see Sheet C)	$7,971 11	
Add amount of Excess Income over Expense incurred for 1904–5 (see Sheet B)	1,450 61	
Add amount of liability to Reserve Fund, Initiation Fees, which was not covered by a cash asset in Savings Bank or cash invested in Engineering Building	206 00	
		9,627 72
		$89,430 45
Less Actual Cost to complete Volume XXV., for 1903–4, over amount reserved for that purpose, *i.e.:*		
Actual cost	$6,837 28	
Amount reserved at September 30, 1904	6,610 00	
Excess charged against Surplus of September 30, 1904 (see Sheet C)		227 28
Surplus, September 30, 1905, as reported (see Sheet B)		$89,203 17
That is:		
Excess Income over Expense incurred 1904–5 (Sheet B)	$1,450 61	
Reserve Fund invested in Engineering Building	7,971 11	
Amount of liability to Reserve Fund, Initiation Fees, written off	206 00	
	$9,627 72	
Less actual amount to complete Volume XXV. over amount reserved at September 30, 1904	227 28	
Net Increase in Surplus	$9,400 44	

The items which have increased the assets at September 30, 1905, over those at September 30, 1904, may be summarized as follows:

Reserve Funds invested in Engineering Building		$7,971 11	
House Furniture—Net additions	$24 20		
Insurance paid in advance	101 20		
Due from Sundry Debtors	54 01		
Increase Stock Second-hand *Transactions*	186 00		
Suspense Account	13 05		
		378 46	
Funds deposited in Savings Bank:			
Trust Fund—M. E. L. A., including interest for 1905	$430 16		
Geo. W. Weeks Legacy Fund, Interest	69 08		
		499 24	
Library—Increased valuation due to Additions		704 15	
Stock A. S. M. E. *Transactions*, increase as per inventory September 30, 1905		1,112 68	
Total Increase in Assets			$10,665 64
Less decrease in assets as follows:			
Heating and Ventilating apparatus, depreciation 1904–5		$291 60	
Stock of badges, decrease in inventory	$58 50		
Arrears of dues	81 67		
Arrears of Funds	33 00		
D. Van Nostrand Company Exchange Account	17 75		
Suspense Account (Over-due)	23 10		
Cash East River National Bank	58 30		
		272 32	
Sundry Debtors (Accounts due Society)		400 41	
Reserve Fund, due to Investment of Funds in Land for Engineering Building		1,601 38	
Total Decrease in Assets			2,565 71
Net Increase in Assets			$8,099 93
Assets at September 30, 1904	$95,433 29		
Add Net Increase as above	8,099 93		
Assets September 30, 1905	$103,533 22		

The items which have decreased the liabilities during the year 1904–5 may be summarized as follows:

Net gain cash asset Reserve Fund (see footnote, Tabular Statement)	$2,339 58	
Altoona Mechanics' Library	1 25	
Reserve for completion of Volume.—Reserve this year less than last	20 00	
		$2,360 83
Less decrease in liabilities:		
Advance payments	$200 25	
Increase in Trust Fund Mech. Eng. Library Association—Receipts and Interest	397 16	
Increase in Geo. W. Weeks Fund—Interest	69 08	
Increase in Library Development Fund—Receipts and Interest	393 33	
Engineering Magazine, credit with Society	50	
		1,060 32
Net Decrease in Liabilities		$1,300 51

Liability at September 30, 1904	$15,630 56
Deduct net Decrease	1,300 51
Liability September 30, 1905	$14,330 05

TRUST, RESERVE AND GEORGE W. WEEKS LEGACY FUNDS.

Liability to Trust Fund Mechanical Engineers' Library Association (including Interest)		$2,335 31	
Cash Asset in Union Dime Savings Institution			$2,335 31
Liability to Library Development Fund, American Society of Mechanical Engineers (including Interest)		1,063 17	
Cash Asset in Union Square Savings Bank			1,063 17
Reserve Funds American Society of Mechanical Engineers, Life Membership and Initiation Fee, Total credited	$9,556 03		
Less Amount of this Reserve Fund that has been invested in Engineering Building during Fiscal Year 1904–5	7,971 11		
Liability Reserve Funds (uninvested)		1,584 92	
Cash Asset in Union Square Savings Bank			1,584 92
Liability to George W. Weeks Legacy Fund, American Society of Mechanical Engineers, including Interest		2,026 08	
Cash Asset in Union Square Savings Bank			2,026 08
Total Liability to Funds		**$7,009 48**	
Total Cash Asset to Funds (Savings Bank)			**$7,009 48**

SHEET B.

COMBINED STATEMENT,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

INCOME AND EXPENSE ACCOUNT, *September* 30, 1905. FISCAL YEAR, 1904–1905.

Income Earned.

Dues Account—		
Dues collected—Fiscal Year 1904–5	$35,194 19	
Outstanding and considered good, Fiscal Year 1904–5	2,800 81	
	$37,995 00	
Less 1 p. c. total cash collections carried to Reserve Fund—A. S. M. E.	381 86	
		$37,613 14
Life Membership	$298 50	
Less—100 p. c. carried to Reserve Fund—A. S. M. E.	298 50	
		0 00
Initiation Fees	$6,050 00	
Less—90 p. c. carried to Reserve Fund—A. S. M. E.	5,445 00	
		605 00
Sales Account (less allowances and goods returned)—		
Publications, except those furnished exchanges for books for the Library	$1,863 50	
Publications — *Transactions* furnished exchanges for books and papers for the Library	488 00	
		2,351 50
Sales of Electros—Net gain		2 34
Sales of Second Hand Vols. Trans.—Net gain		60 00
House Account, Rent—		
For Sleeping Rooms	$815 50	
" Hall	55 00	
		870 50
Increase Stock Transactions on Hand—		
Inventory, Sept. 30, 1905	$16,105 66	
Less— " " 1904	14,992 98	
		1,112 68
		$42,615 16

Expenses Incurred.

Transactions—			
Paid on account of Volume XXVI.—Fiscal Year 1904–5		$7,440 88	
Add—Estimated amount required to complete Vol. XXVI. for Fiscal Year 1904–5, including cost of distribution		6,590 00	
			$14,030 88
New Edition Volume XXIV. Transactions			740 00
Office Account—			
Expenses other than Salaries—			
Stationery and Printing		$564 67	
Postage, general		641 02	
Telegraph, Telephone and Messenger		89 13	
Supplies		319 04	
Incidentals		231 05	
		$1,844 91	
Salaries		10,200 00	
			12,044 91
Meetings—			
Annual		$572 07	
Spring		597 00	
Monthly		244 33	
			1,413 40
Committee Work—			20 00
Year Book and Pocket-Lists of Members—			
Year Book—Jan., 1905, Composition, Presswork and Paper		$1,707 53	
Pocket List—July, 1905, Composition, Press Work and Paper		922 81	
Distribution on both—postage and envel.		347 41	
			2,977 75
Circulars, including distribution—			
Admission		$983 25	
Meetings		634 70	
General		371 51	
			1,989 46
Library—			
Binding Books		$287 20	
Expenses		278 40	
Salary, Librarian		600 00	
			1,165 60
Certificates and Membership Cards—			
Certificates		$234 79	
Cards		18 84	
			253 63
Distribution and Repair of Badges—			
Repairs		$5 35	
Distribution		33 98	
			39 33
Legal Expenses and Expert Fees—			
Legal Expenses		0 00	
Expert Fees, Auditing Books		$105 00	
			105 00
Uncollectable Accounts—Written off			26 15
House Account—			
Lighting	$725 69		
Fuel	367 70		
Janitor's Supplies	220 50		
Laundry	407 62		
Insurance	80 30		
Wages	1,773 00		
Incidentals	193 46		
Total Cost of Operating House		$3,768 27	
Interest on Mortgage of $33,000 @ 4¼ p. c.		1,402 50	
Repairs and Renewals—			
House	$249 85		
Furniture	375 68		
		625 53	
Depreciations—			
House Furniture	$136 17		
Heating and Vent. App.	291 60		
		427 77	
Total Expense of House exclusive of Interest on Value of Equity			6,224 07
Headquarters, St. Louis Exposition			134 37
By Balance—Excess Current Income over Expense incurred for Fiscal Year 1904–5, credited to Surplus account			1,450 61
			$42,615 16

Certified to as correct

FRANCIS W. HOADLEY,

Cashier and Assistant to Secretary and Treasurer.

SHEET C.

COMBINED BALANCE SHEET,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION.

September 30, 1905.

Assets.

Property—12 W. 31st Street, N. Y. Value Appraised		$85,000 00	
Less Mortgage		33,000 00	
			$52,000 00
Fixtures and Furniture—Heating and Ventilating apparatus. Book Value, Oct. 1, 1904	$2,916 00		
Less 10 p. c. Depreciation for 1904–5	291 60		
Book Value, Sept. 30, 1905		2,624 40	
House Furniture—Book Value, Oct 1, 1904	$1,361 72		
Less 10 p. c. Depreciation for 1904–5	136 17		
	$1,225 55		
Add for New Furniture Purchased 1904–5	160 37		
Book Value, Sept. 30, 1905		1,385 92	
			4,010 32
Library—Books, Pamphlets, etc.—			
Book Value, Oct. 1, 1904		$11,884 52	
Additions during year		704 15	
			12,588 67
Stock of Transactions—Inv. Sept. 30, 1905			16,105 66
Stock of Badges—Inv. Sept. 30, 1905			94 50
Arrears of Dues—Fiscal Year 1904–5			2,800 81
Arrears of Trust Funds—Fiscal Year 1904–5			....
Insurance Premiums Paid in Advance			188 66
Sundry Debtors—Due the Society:			
For Volumes and Pamphlets		$58 74	
" Room Rent		68 00	
" Badges		6 50	
" Miscellaneous		21 26	
			154 50
Suspense Account (over-due accounts)—			
Room Rent		$13 50	
Publications and Pamphlets		17 15	
			30 65
D. Van Nostrand Co.—Exchange Account:			224 00
Spon & Chamberlain—Balance due the Society for Publications.			16 50
Stock Second-hand A. S. M. E. Transactions—Inv. Sept. 30, 1905			312 00
Investment Engineering Building—From Reserve Fund, A. S. M. E.		$7,971 11	7,971 11
Union Dime Savings Institution, New York Trust Fund, M. E. L. A.—On deposit—			
Principal	$2,249 23		
Interest	86 08		
		$2,335 31	
Union Square Savings Bank, New York Reserve Fund, A. S. M. E.			
Life Membership—Principal	18 53		
" " Interest	.....		
		18 53	
Library Devl. Fund—Principal	$1,046 12		
" " " Interest	17 05		
		1,063 17	
Initiation Fees—Principal	$1,566 39		
" " Interest	....		
		1,566 39	
Total Reserve Fund		$4,983 40	
Geo. W. Weeks Legacy Fund—			
Principal	$1,957 00		
Interest	69 08	2,026 08	
Total Funds in Savings Banks			7,009 48
Cash—East River National Bank			26 36
			$103,533 22

Liabilities.

Reserved for Uncompleted Work on Volume XXVI. for Year 1905–6—			
Composition and Electrotyping		$1,800 00	
Binding—Paper covers		75 00	
" Leather		2,700 00	
Boxing Plates		15 00	
Distribution—Postage and Express		1,200 00	
Revised Papers		800 00	
			$6,590 00
Reserved for Interest on Mortgage Accrued			350 62
Advance Payments			362 70
*Library Development Fund—			
Principal	$1,046 12		
Interest	17 05		
		$1,063 17	
*Reserve Fund Life Membership Credit, Oct. 1, 1904	$1,007 00		
Add Receipts, 1904–5	298 50		
	$1,305 50		
Add Interest 1904–5	24 87		
	$1,330 37		
Less Amount Invested in Engineering Building, 1904–5.	1,311 84		
		18 53	
*Reserve Fund Initiation Fees Credit, Oct. 1, 1904	$2,917 50		
Add Net Credit for 1904–5	5,239 00		
	$8,156 50		
Add Interest for 1904–5	69 16		
	$8,225 66		
Less Amount of Said Fund Invested in Engineering Building	6,659 27		
		1,566 39	
*Trust Funds, M. E. L. A.—			
Principal	$2,249 23		
Interest	86 08		
		2,335 31	4,983 40
*Geo. W. Weeks Legacy Fund—			
Principal		$1,957 00	
Interest		69 08	
			2,026 08
Altoona Mech. Library—Credit Balance with Society			16 75
Engineering Magazine—Credit Balance with Society			50
Surplus, Sept. 30, 1904		$79,802 73	
Add Reserve Funds Invested in Engineering Building—			
Life Membership	$1,311 84		
Initiation Fees	6,659 27		
		7,971 11	
Add Excess of Current Income over Expenses incurred for Fiscal Year 1904–5 (see Sheet B)		1,450 61	
Add—Amount of Credit to Reserve Fund, Initiation Fees for 1904–5, which could not be covered by a cash deposit in Savings Bank, or invested in Engineering Building		206 00	
		$89,430 45	
Deduct—Excess Actual Expenditure to Complete Vol. XXV. for Year 1903–4:			
Actual Expenditure	$6,837 28		
Reserved to complete volume at Sept. 30, 1904	6,610 00		
		227 28	
Surplus, Sept. 30, 1905			89,203 17
			$103,533 22

* Covered in full by cash in Savings Bank (see Assets).

Certified to as correct.
FRANCIS W. HOADLEY,
Cashier and Assistant to Secretary and Treasurer.

Having audited the above Balance Sheet and accompanying Accounts, prepared by Mr. Francis W. Hoadley, Cashier of the Society, with the books and vouchers of the Society, we certify that the same, in our opinion, fully and fairly represents the condition of the Society at September 30, 1905.

THE AUDIT COMPANY OF NEW YORK.

NEW YORK.

EDWARD T. PERINE,
General Manager and Treasurer.

SHEET D.

TABULAR STATEMENT OF CHANGES IN ASSETS AND LIABILITIES,

AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND MECHANICAL ENGINEERS' LIBRARY ASSOCIATION COMBINED.

September 30, 1905, compared with September 30, 1904,
and remarks in comment thereon.

Designed and compiled by FRANCIS W. HOADLEY, Cashier and Assistant to Treasurer. Showing whence arise the gains and losses in the financial condition of the Society, as affecting its Assets and Liabilities.

Accounts.	Assets and Liabilities. September 30, 1904. Assets.	Total.	September 30, 1905. Assets.	Total.	Losses and Gains. During Year 1904-5. Losses.	Gains.
Property, 12 W. 31st Street, Equity	$52,000 00		$52,000 00			
Fixtures and Furniture—						
Heating and Ventilating Apparatus, Book Value	2,916 00		2,624 40		*$291 60	
House Furniture, Book Value	1,361 72		1,385 92			†$24 20
Library—Books, Pamphlets, etc., Book Value	11,884 52		12,588 67			704 15
Stock of *Transactions* on hand, Inventory Value	14,992 98		16,105 66			1,112 68
Stock of Badges on hand " "	153 00		94 50		58 50	
Arrears of Dues	2,882 48		‡2,800 81		81 67	
" " Sinking Fund	23 00				23 00	
" " Fellowship Fund	10 00				10 00	
Insurance Paid in Advance	87 46		188 66			101 20
Due from Sundry Debtors—Publications	415 00		58 74		356 26	
" " " " Room Rent	36 50		68 00			31 50
" " " " Badges	4 50		6 50			2 00
" " " " Electros	44 15				44 15	
" " " " Miscellaneous	75		21 26			20 51
D. Van Nostrand Co., Exchange Acct., Balance due Society	241 75		224 00		17 75	
Spon & Chamberlain, Balance due Society	16 50		16 50			
Stock Second-hand *Transactions* on hand	126 00		312 00			186 00
Suspense Account (Overdue)—Room Rent	2 00		13 50			11 50
" " " Publications	15 60		17 15			1 55
" " " Electros of Cuts	23 10				23 10	
Investment Engineering Building—Investment Reserve Fund			§7,971 11			§7,971 11
Union Dime Savings Institution, New York, N. Y.—Trust Funds M. E. L. A. on deposit	1,905 15		‖2,335 31			430 16
Union Square Savings Bank, New York, N. Y.—Reserve Funds A. S. M. E. on deposit	4,249 47		‖2,648 09		1,601 38	
Union Square Savings Bank, New York, N. Y.—Geo. W. Weeks Legacy	1,957 00		‖2,026 08			69 08
Cash, East River National Bank, N. Y.	84 66		26 36		58 30	
Total		$95,433 29		$103,533 22		
	Liabilities.		Liabilities.			
Reserved for Interest on Mortgage Accrued	$350 62		$350 62			
Advance Payments	162 45		362 70		200 25	
Trust Funds, Mechanical Engineers' Library Association—						
Fellowship	373 32		443 37		70 05	
Sinking	1,564 83		1,891 94		327 11	
Altoona Mechs. Library—Credit Balance with Society	18 00		16 75			1 25
Geo. W. Weeks Legacy	1,957 00		2,026 08		69 08	
Reserved for Uncompleted Work, Volume of *Transactions*	¶6,610 00		6,590 00			20 00
Library Development Fund	669 84		1,063 17		393 33	
Reserve Fund, A. S. M. E.—Life Membership	1,007 00		18 53			ɸ988 47
" " " Initiation Fees	2,917 50		1,566 39			♯1,351 11
Engineering Magazine—Balance due them			50		50	
Total		15,630 56		14,330 05		
Surplus, September 30, 1905				$89,203 17		
" " " 1904		$79,802 73		79,802 73		
Total Increase						$13,026 47
" Decrease					$3,626 03	3,626 03
Net Increase in Assets:						
+ Reserve Funds invested in Engineering Building . . $7,971 11						
+ Excess of Current Income over Expenses incurred for Fiscal Year 1904-5 . . 1,450 61						
+ Amount of Liability to Reserve Fund—Initiation Fee Receipts, 1904-5—which was not covered by either Cash Asset in Savings Bank or Cash Investment in Engineering Building . . 206 00						
$9,627 72						
— Excess Actual Expenditure to complete Volume XXV. over amount reserved for same . . $227 28						
				$9,400 44		$9,400 44

* 10 per cent. depreciation written off.

† 10 per cent. depreciation written off, and amount expended for new furniture added.

‡ All dues for the year 1904-5, and considered as collectable.

§ Actual cash from Reserve Fund invested in Engineering Building. Full payment our assessment for 1904-5 to the United Engineering Society, the holding Society for the Engineering Building.

‖ A total cash asset of $7,009.48 has been created in the last three years against a total liability to funds of $7,009.48, and in addition to this cash asset we have invested from Reserve Fund cash the amount of $7,971.11 in the Engineering Building, making the total asset accumulated $14,980.59, this having been made possible largely by the great number of new members elected and the initiation fees received from them.

¶ Amount actually paid to complete Vol. XXV. was $6,837.28—that is, the amount reserved was $227.28 too small, excess being charged against Surplus at September 30, 1904 (see Sheet C).

ɸ This gain due to the following—

Liability, September 30, 1904, as shown above		$1,007 00	
Add Interest for 1904-5	$24 87		
Cash receipts, 1904-5	298 50	323 37	
			$1,330 37
Less Cash Asset in Savings Bank (see Sheet C)			18 53
Actual Cash invested in Engineering Building 1904-5 (see Sheet C)			$1,311 84
Less total increase for 1904-5, as above			323 37
Net gain as above			$988 47

♯ This gain due to the following—

Liability, September 30, 1904, as shown above		$2,917 50	
Add Interest for 1904-5	$69 16		
Reserve, 1904-5	5,239 00	5,308 16	
			8,225 66
Less Cash Asset in Savings Bank (see Sheet C)			1,566 39
Actual Cash invested in Engineering Building 1904-5 (see Sheet C)			$6,659 27
Less total increase for 1904-5, as above			5,308 16
Net gain, as above			$1,351 11

PAYMENT ON LAND FOR NEW ENGINEERING BUILDING.

Payment First Assessment, 1904–5.

First Payment, 1904–5	$7,971 11	First Assessment, 1904–5	$8,000 00
Interest rebate equalization of payments	28 89		
	$8,000 00		$8,000 00

Second Assessment, 1905–6.

Cash on hand in Savings Banks that can be applied to meet Second Payment:		Second Assessment, to be met by July 1, 1906	$8,000 00
A. S. M. E. Reserve Fund	$1,585 00		
M. E. L. A. Fellowship Fund	2,335 00		
	$3,920 00		
Amount to be raised to balance	4,080 00		
	$8,000 00		$8,000 00

At the close of the formal presentation of this report the Secretary read the Report of the Tellers of Election of Members, as follows:

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as Tellers under By-Laws 6, 7, and 8, to scrutinize and count the ballots cast for and against the candidates proposed for membership in their several grades, in the American Society of Mechanical Engineers, and seeking election before the 52nd Meeting, New York, N. Y., 1905.

They have met upon the designated days in the office of the Society and have proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society to the election of the following persons, whose names appear on the appended list in their several grades.

There were 693 votes cast on the ballot ending November 27, 1905, of which 13 were thrown out on account of informalities.

The Tellers have considered a ballot as informal which was not endorsed, or where the endorsement was made by a facsimile or other stamp.

CHAS. E. LUCKE,
H. G. CHATAIN,
ALBERT SPIES. } *Tellers of Election.*

MEMBERS.

Antisel, F. L.
Baker, C. W.
Baush, G. H.
Blood, L. H.
Brakley, W. J.
Buttolph, B. G.
Chester, J. N.
Crawford, C. W.
Crook, G. L.
Daugherty, S. B.
Dennison, W. N.
Eaton, C. E.
French, E. V.
Guldlin, O. N.
Guy, A. E.
Hamilton, W. J.
Hansen, T. H. C.
Harris, J. W.
Haskins, C. D.
Hopkins, G.
Hopps, J. H.
Kelley, F. W.
Larsson, T. L. F.
Ludy, L. V.
McClellan, W.
Maytham, W. J.
Mersereau, T. T.
Moritz, A.
Newton, P. A.
Penney, R. C.
Pettis, C. D.
Proal, A. B., Jr.
Riddle, H. S.
Scarborough, F. W.
Seymour, D. S.
Snow, W. G.
Strom, C. A.
Suck, A.
Taubenheim, U. E.
Thomas, J. W.
Thompson, D.
Tuttle, W. B.
Wheeler, W. T.

Promotion to Full Membership.

Alexander, C. A.	Gray, J. L.	Marr, G. H.
Cory, H. T.	Guelbaum, D.	Mitchell, B. M.
Faig, J. T.	Jefferies, F. L.	Rippey, S. H.
Farrand, D.	Larkin, A. C.	Von Ammon, S.
Gnade, E. R.	McClintock, E. H.	Weber, O. L. E.
	Mackintosh, F.	

Associates.

Bendit, L.	Kilgour, D. F.	Salter, T, F.
Graves, J. M.	Knight, G. L.	Sandford, W. E.
Hofmeyer, G. A.	Parker, J. C.	Shaw, A. D.

Promotion to Associate.

Cole, E. S.	Krebs, A. S.	Pitkin, J. L.
Dow, C. S.	Libby, M. M.	Young, W. A.
Gunther, C. O.	Moody, H. A.	

Juniors.

Alexander, A. T.	Fallon, J. B., Jr.	Miller, T. H.
Alexander, L. B.	Farmer, T., Jr.	Parish, W. H.
Allen, C. B.	Frecker, A. N.	Pope, H. F.
Appleton, W. D.	Gamble, W. H.	Reed, E. H.
Arnold, Otto, Jr.	Gillies, W. F.	Robinson, L. G.
Aull, J. J.	Glenn, C. S.	Royle, V. E.
Bachelder, J. E.	Goentner, W. B.	Sample, M. de F.
Barnes, E. A.	Hagerty, W. W.	Seaman, E. H.
Berryma.	Hall, M. A.	Soper, E. C.
Borden, W. H.	Heineken, W. P.	Stewart, G. W.
Bright, H. De H.	Herb, A.	Symonds, N. G.
Cazenove, L. A. de.	Hill, E. G.	Sze, S. Z. T.
Chambers, N. C.	Karr, E. M.	Turner, C. H.
Chandler, S. McK.	Kevorkian, Z. H.	Van Deinse, A. F.
Coward, H.	Klahr, C. D.	Watson, G. L.
Cressler, G. H.	Koon, S. G.	Wear, B. C.
Crothers, Chas. E., Jr.	Kniskern, W. H.	Weinland, H. G.
Davidson, J. B.	Kothny, G. L.	Wilcox, C. C.
Dreyfus, E. D.	Masury, A. F.	Williams, E. D.
Eayrs, T. C.	Miller, F. J.	Wisewell, F. H.

Following this, the Report of the Tellers for Officers for the Society year about to open, was presented and read as follows:

REPORT OF TELLERS.

The Committee of Tellers appointed to count the ballots cast by the members for officers of the American Society of Mechanical

Engineers for the year 1905-1906, begs to submit the following report:

Total ballots cast	869
Ballots thrown out unsigned	15
Total ballots counted by tellers	854

Of the regular ballots counted by the Tellers, they would report the following result:

For President.

Fred. W. Taylor, Philadelphia	833
Scattering	6

For Vice-Presidents.

Walter M. McFarland, Pittsburg	834
Edward N. Trump, Syracuse	830
Robert C. McKinney, New York City	830
Scattering	3

For Managers.

Walter Laidlaw, Cincinnati	831
Frank G. Tallman, Cleveland	826
Fredk. M. Prescott, Milwaukee	831
Scattering	1

For Treasurer.

Wm. H. Wiley, New York City	836
Scattering	1

Our count shows therefore election of

Fred. W. Taylor	*President.*
Walter M. McFarland Edward N. Trump Robert C. McKinney	*Vice-Presidents.*
Walter Laidlaw Frank G. Tallman Fredk. M. Prescott	*Managers.*
Wm. H. Wiley	*Treasurer.*

Respectfully submitted,

H. F. J. Porter,
A. L. Colby,
Chas. R. Pratt. } *Tellers of Election.*

Pursuant to the accepted custom in the Society, Messrs. John Fritz and S. T. Wellman were designated a Committee to escort the President-elect, Mr. Fred. W. Taylor, to the front of the room where he was welcomed by the President, to which Mr. Taylor made a fitting response.

The Chair then called upon the Committee representing the Society for the construction of the Engineering Building. This report was presented by Mr. Charles Wallace Hunt, of the committee as follows:

In reporting for the Committee representing this Society in planning and working out the details of the Engineering Building, there is little to say beyond a somewhat formal report of progress. The ground was all new to the Committee and to the donor himself when we began; Mr. Carnegie simply gave the money in the form of an agreement to pay bills, but we had to encounter delays in securing suitable property and in perfecting the required organization, all of which could not be talked about in public for obvious reasons.

Then came the problem of design of the building. It was not a purely office structure, so that we could not copy experience in those lines. It was not a club building, but it was a new departure in society headquarters. There was also a considerable work in harmonizing different ideas, before the architects' work could be brought to suit all interests. We have distributed to all members a pamphlet showing the details of the important floors. I call special attention to the large auditorium surrounded by a corridor, so that conversation and other noises which have been so much in evidence this morning shall not disturb the sessions within.

After the plans had been settled there was the necessary consideration and necessary modification of the bids resulting from the changes in material and it was not until July, 1905, that the contract was let. Work is now in progress and moving rapidly. Unless something which we do not know or foresee should arise, we have hopes that the next annual meeting can be held in the new building. While strikes are probably inevitable, it is possible that our building may not be included in the list of those affected.

I want particularly to call the attention of members to the opportunity and need of developing the Society's library; we have not been making progress in this respect during these last few years for reasons connected with the financing of the Society, and with the consideration of plans looking to a consolidation. The Mining

Engineers have a particularly fine library of reports which they are rapidly augmenting at the present time. The Electrical Engineers have the Latimer Clarke Library, which is probably the finest of its class in the country, and they are making rapid progress in procuring new books.

The Committee feel that this Society should take some steps so that the Library which results from bringing the three collections together shall be at least equal on the mechanical side to what the others bring in their special lines. If there are any further questions to be asked I should be glad to answer them.

At the close of this report the Chair asked from Mr. H. H. Suplee a memorandum of progress concerning the work of the Historical Committee.

Mr. Suplee reported that the data of the Committee had been very largely gathered together from correspondence and personal reminiscences, and for this reason the work had been much slower than they had desired or expected.

The purpose had been to celebrate the twenty-fifth anniversary of the Society's formation by the issue of this historical record, but it would reach the members sometime during the approaching year.

The Chair then called for a report of the Society's professional committee on a Proposed Standard for Machine Screws. This was presented by Mr. George M. Bond, of the Committee, and received discussion from Messrs. L. D. Burlingame, E. O. Goss, G. A. Gulowsen, S. A. Moss, and John W. Upp.

At the close of the discussion Mr. Bond stated that the Committee held this report as a preliminary document and welcomed contributions for consideration, and that such contributions received in proper season would be published as part of the discussion and considered by the committee in the preparation of the final report, which they hoped to present before the meeting of the Society in the spring.

The Chair called for the presentation of any general business, but none being offered professional papers were taken up until the hour for adjournment.

The paper of the morning was that of Mr. Jay M. Whitham, entitled "Use of Natural Gas Under Boilers." It was discussed by Messrs. E. G. Bailey, J. R. Brown, W. F. M. Goss, E. A. Hitchcock, Wm. Kent, and S. T. Wellman.

At the close of this discussion the meeting adjourned until the evening.

THIRD SESSION. WEDNESDAY EVENING, DECEMBER 6TH, 8.30 O'CLOCK.

The Meetings Committee in arranging its program had invited for this session Professor R. W. Wood, of Johns Hopkins, to present an illustrated lecture to describe and explain some phenomena capable of being caught by the sensitive photographic plate, but which ordinarily the eye could not work quickly enough to see. The lecture was illustrated both with lantern slides and with cinematograph films, and was greatly enjoyed.

FOURTH SESSION. THURSDAY MORNING, DECEMBER 7TH, 10.30 O'CLOCK.

The program for this morning called for a discussion to cover the general subject of Bearings, and other invited topics. The Committee had divided the subject into the following sub-headings:

1. Metals suitable for bearings.
2. Lubrication of bearings.
3. Method of cooling bearings.
4. Limits of speeds and pressures.
5. Designs of bearings for high speeds and high pressures.
6. Thickness of oil film or allowance between journal and bearing.
7. Thrust bearings.
8. Ball bearings.
9. Roller bearings.

Those who took part in the discussion were Messrs. Geo. M. Basford, P. H. Been, R. C. Carpenter, G. W. Dickie, S. S. Eveland, G. R. Henderson, Henry Hess, A. H. Johnston, H. K. Jones, A. Kingsbury, A. M. Mattice, F. Mossberg, C. W. Naylor, Chas. R. Pratt, H. G. Reist, W. S. Rogers, Oberlin Smith, H. H. Suplee, F. W. Taylor, John W. Upp, and J. J. White.

The interest attaching to this discussion was such that a very general sentiment prevailed that this method was one which could properly be developed with great advantage to the Society and its members. At the close of the discussion on Bearings the paper on "Reinforced Concrete Applied to Modern Shop Construction," by Mr. E. N. Hunting, was presented and discussed. The subject of fire protection naturally allied itself to the concrete design of

buildings, and Mr. H. F. J. Porter presented by invitation some recent research upon the question of the safety of factory employees.

FIFTH SESSION. FRIDAY MORNING, DECEMBER 8TH, 10 O'CLOCK.

The session was called promptly to order for the discussion of the papers submitted by authors and accepted by the committee for reading and presentation.

The titles of the papers were as follows: R. J. Durley, "Measurement of Air Flowing Through Circular Orifices in Thin Plates"; R. H. Fernald, "Results of Preliminary Producer Gas Tests," United States Geological Survey Testing Plant, St. Louis, Mo.; Charles E. Lucke, "Pressure Drop Through Poppet Valves;" A. J. Herschmann, "Test of Elevator Plant," Trinity Building, New York city; H. F. J. Porter, "The Realization of Ideals in Industrial Engineering."

Those who took part in the discussion were Messrs. R. P. Bolton, Thos. R. Brown, Wm. H. Bryan, John Calder, A. A. Cary, Hugo Diemer, John T. Hawkins, Geo. Hill, F. E. Junge, Albert Kingsbury, R. E. Mathot, E. S. Matthews, S. A. Moss, John C. Parker, Chas. R. Pratt, W. S. Rogers, W. B. Snow, H. H. Suplee, and S. S. Wyer.

At the close of the discussion the Secretary called attention to the fact that the handsome monogram in electric lights was a gift to the Society, and on motion a vote of thanks was extended for this gift and for the other courtesies which the Society had enjoyed during the continuance of the meeting. The President asked Mr. Fred. W. Taylor to come to the platform and turned over to him the responsibility of adjourning the meeting. Mr. Taylor spoke of the problem before the President of choosing members of the Society to supply vacancies on the standing committees and the relative advantages of two differing policies to determine the choice. On the one hand was the advantage of making the committees territorially representative of the Society by choosing members qualified to serve, from a wide geographical distribution, so that they should be representative not only in their own persons, but representative also of the widely distributed interest of the members by residence.

The objection to this policy was the difficulty of securing an effective quorum for the transaction of business when committees

were required to act on short notice or at short intervals when the importance of the individual items might perhaps not be large, and yet the necessity for prompt action insistent.

The other policy was to choose the members for the committees from within relatively short radius of the Society's headquarters, so that effective attendance could be secured for the transaction of business. The objection to this system was the obvious one that the management of the Society through its committees appeared to concentrate itself rather than to secure a wide distribution. The President asked the meeting to express its opinion as to which of the two policies the best interests of the Society required that he should follow. It was, on motion,

Resolved, That it was the sense of the meeting that the best interests of the Society demanded the selection of the members of standing committees from those whose residences would enable them to be present at the meetings and who could be promptly summoned for special meetings when such were necessary. This resolution, duly put by the Chair, was carried.

The Chair then put the motion to adjourn, which was duly seconded and carried.

The expected place of the next meeting of the Society is the city of Chattanooga, Tenn.

On the afternoon of Wednesday the members were the guests of the new Henry R. Worthington Hydraulic Works at Harrison, N. J. A special train from the terminal of the Delaware, Lackawanna, and Western Railway carried the members to the yard, and after being photographed they were entertained at luncheon and distributed, under the guidance of officers of the Company, for a visit to the extensive plant. The numbers in attendance on this excursion were phenomenal to the point of embarrassment, as over 870 persons were on the train and others joined the party at the works. A brief address by Mr. William Schwanhausser, member of the Society and General Manager of the Company, could scarcely be heard by the large number to be reached.

On the afternoon of Thursday the members were the guests of the New York School of Automobile Engineers at 146 West Fifty-sixth Street, to inspect the plans which had been gotten together by those interested, for the instruction of operators for motor vehicles. The principle of the scheme of instruction was to apply to trade teaching the methods of the mechanical laboratory, so

that the students should learn the principles of the various operations which they were expected to control rather than simply to learn how to perform certain designated operations on a specific apparatus.

The members were also invited to visit the Watson-Stillman Company at Aldene, N. J., and the Waterside Power Station of the Edison Company, on the east side of the city.

No. 1096.*

ON THE SAFEGUARDING OF LIFE IN THEATERS.

A STUDY FROM THE STANDPOINT OF AN ENGINEER.

BY JOHN R. FREEMAN, PROVIDENCE, R. I.
(Member of the Society.)

Custom has decreed that the President of this Society should choose his subject for the opening address of our winter meeting from within some field of his own special work, and that the address should be either a historical review, or an effort to lead the thought of the evening into some useful line of advance in applied science; and so I bring to you a topic that has been much in my thought for two years past, and in one corner of the field of fire protection, to which I have devoted a portion of my time for twenty years.

It is a fair and moderate statement that the present practice of the art of fire prevention, as applied to theaters and buildings of public congregation, is from ten to twenty years behind the fire protection of the best industrial works, and true that the fire hazard to theater property in general, as measured by a comparison of insurance rates, is ten to twenty times as great for the modern theater as for the modern factory.†

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* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

† At the time of the Iroquois fire the average cost of insurance per year on the principal Chicago theaters was, on buildings 3.7 per cent., on furnishings and fixtures 4.3 per cent., on scenery 4.7 per cent.

On the best fireproof theater buildings in Chicago it was about 1 per cent., with 2 per cent. on fixtures, furnishings, and scenery therein. On some of the more hazardous theater structures in Chicago the rates were 6 per cent. and even 7 per cent. per year. The same insurance companies that insure these theaters will insure a strictly first-class cotton mill, or even a first-class woodworking or rubber factory, at $\frac{1}{10}$ to $\frac{1}{5}$ of 1 per cent. per year, when thoroughly protected by automatic sprinklers, etc.

We must bear in mind that a comparison of insurance rates, while an excellent guide, is not a complete or accurate basis for a comparison of safety to life in

All of this is unnecessary. It is a wrong against the public that should be righted. The actual fire hazard at the theater can be made smaller than that of the factory by well-proved means, the cost of which is not extravagant. The safeguards needed are mostly simple; the main features of some of them are already worked out and well proved within the great factories which you engineers build and manage; the additional safeguards required to be worked out, or adjusted for this special case—the automatic smoke vents—the safe proscenium curtain—the safe warming and ventilation—the proper arrangement of automatic sprinklers in stage and dressing-rooms and storerooms, are within the field of the Mechanical Engineer, and are mostly simple problems when serious attention and skill are once directed to them.

As a society of engineers, we have a precedent for giving our time to this study in the investigation made by the Austrian Society of Engineers after the burning of the Ring Theater in Vienna, and republished by them after the burning of the Iroquois.

In the great factories of New England first, and more recently in those of the Middle States and Middle West, all represented largely in our membership, there have been slowly worked out the most advanced methods of fire prevention that are anywhere to be found. This safety of the slow-burning American factory has come first through an appreciation of the danger and then a study by one engineer after another of how to meet it; then a conscientious attention to perfection of detail, and then an education of the average workman about the place into the requirements for safety.

In the course of my own studies of the theater and auditorium problem, I have seen almost everywhere conditions affecting the safety of life that would not be tolerated by the managers of our best industrial works, and all from simple failure to know or to give attention.

For example, I have seen in one of the best New York theaters the wedge-shaped space beneath the sloping floor of the auditorium used as a storeroom for trunks and properties. This room was also the plenum chamber for the ventilation. Suppose that rats and matches, spontaneous ignition of oily material, or any of

different theaters, for the questions of accident or death to audience and actors are mostly settled within the first five minutes after fire breaks out, while the per cent. of damage, that concerns the fire underwriter, may be in suspense for an hour or more.

the obscure but frequent causes should start even a slow, smouldering fire in this room. Why is it not foreseen that the smoke rising through the air ducts in the floor might throw the audience into a panic and cause great loss of life?

In one of the most famous halls in America I found the portable wooden flooring, used sometimes to level up and transform the main seating space into a ballroom, stored in a dark passageway, which formed the main air chamber between the heating coils and the concert hall, all thus kiln-dried to perfection, and when I showed it to the manager and to an intelligent aldermanic committee and urged its immediate removal, they saw no danger and thought me hypercritical, and could not even see that automatic sprinklers would be of use in such a concealed storage space.

In Chicago, within a few months after the appalling disaster at the Iroquois Theater, the aldermen rescinded the rule calling for automatic sprinklers over the stages and rigging lofts * of the theaters because the managers believed they "wouldn't do any good," and "might start a panic should one happen to open prematurely." Every factory manager or mill engineer in this audience will admit the absurdity of such a statement.

In Boston, the law still accepts the non-automatic sprinkler pipe to be opened by hand, a device which has now been almost totally discarded in factory fire protection in favor of the automatic.

Most dangerous of all, I have found behind the scenes and in the mechanics' rooms a lack of the scrupulous neatness and order that characterizes a modern, well-organized factory; have found a multitude of dark, concealed spaces used as catch-alls, and an apparent lack of appreciation by owner and architect that *a flood of daylight in storerooms, workrooms and dressing-rooms is the best of all safeguards*, by making dirt, disorder and dangerous rubbish conspicuous. While there are notable exceptions, the atmosphere of the theater is largely of show and tinsel, and this contributes to the less thoroughgoing standards of neatness and completeness than in the factory.

We cannot leave it to the underwriter to make the theater safe against fire. The able president of one of the largest insurance companies has said to me, "As an individual, I would be very glad to see the theaters safe for the public which patronizes them, but

* They are insisted on in the mechanics' rooms, and in other places far less dangerous to the audience.

as an underwriter *I charge for the hazard as I find it*, and need not care particularly whether the rate is one per cent. or five per cent." He tells me, too, that on the whole the theater class at current rates is profitable underwriting.

We cannot leave it with the framing of a good building law. The same underwriter also said to me, "The City New York has a pretty good building law, yet the city is full of theaters that are unsafe, some of them constructed since the building law went into effect." The Chicago Building Law required automatic sprinklers over the stage; until after the Iroquois, not one had ever been put in. Then, in the effort to perfect the enforcement of the law, they cut out its requirement for sprinklers over the stage!

How can we transfer the care and the precautions of the modern factory to the modern theater? How can we bring the manager, the architect, and the official guardians of public safety—the fire chiefs and the public inspectors of buildings—to understand and introduce the well-proved safeguards, and to be critical about that perfection of detail on which safety depends? How can we bring the public to demand these things?

Our fellow-member, Mr. Gerhard, presented some of these matters admirably some years ago in a series of popular talks which he recast into a most useful and suggestive little book on Theater Fires.

A German engineer, Herr August Foelsch, of Hamburg and Vienna, began in 1869 to collect statistics of theater fires, and up to the time of his death had collected records of over 500. This list has been extended by Mr. E. O. Sachs, a London architect, until it contains some account of 1,000 theater fires that have happened in various parts of the world within about 100 years. The American engineer, Hexamer, has also added useful contributions to this record. These figures are impressive, but *they teach far less than a full study of a few of the notable examples.*

The Example of the Iroquois.

I first became actively interested in this question by the burning of the Iroquois Theater at Chicago a little less than two years ago. A prominent manufacturer, two of whose little nieces were among the nearly 600 people that perished, wired me to come over to Chicago and investigate; in a noble spirit he said, not for the

purpose of fixing the blame, but to help us find out how such fearful disasters can be prevented.

I examined the structure before any of the wreckage had been moved, listened to evidence before the coroner's inquest, counseled with the mayor and committee of the Board of Aldermen, questioned eye-witnesses, visited Chicago repeatedly, and for several months devoted to this study all of the time that I could get release from business, and inspected many other theaters in the effort to reach a clearer understanding of their special hazards.

This fire at the Iroquois Theater occurred at a Wednesday afternoon matinee, in the midst of the holiday season, when the theatre was crowded, largely with pleasure parties of women and children.

A spectacular play was being given; the amount of scenery was uncommonly large; the fire was caused by a spark from a portable electric arc light, known as "spot light"—used to throw a strong light on a special group—which set fire to one of the draperies. The fire spread in the hanging sheets of scenery with great rapidity and it is probable that in from one to two minutes the great mass of scenery on the stage was in flames. Meanwhile an unsuccessful attempt was made to lower the asbestos curtain—the leading comedian came forward and urged the audience to keep their seats. A door, opened by the escaping actors, let a great rush of air inward—this together with the expansion of the air in the top of the stage space by the heat drove the flames out under the proscenium arch into the upper part of the auditorium. Here was instant discovery—cool, prompt action by the theater staff. There was, perhaps, a momentary delay in sounding the public fire alarm, but with admirable promptness the chief of the public Fire Department and an efficient force of firemen were on the ground within little more than five minutes from the first alarm—we can never hope for prompter or better service from a public fire department—but even by that short time *most of the victims had already become suffocated.*

Some of the cooler headed, who followed the maxim for safety, "Remain in your seat and avoid crushing at the exit," were suffocated in the gallery where they sat.

Out of an audience of about 1,830, there were 581 killed, or 32 per cent., and it is said about 250 more were injured.

Of those killed, about 400 occupied the gallery, or 70 per cent. of those in the gallery perished; and about 125 occupied the balcony,

or 30 per cent. of those in the balcony perished. Of those who occupied the floor not more than 7 were killed, and most of these deaths, it is said, were caused by persons jumping from the gallery.

Suffocation was the main cause of death. The underwriters' loss was small as theater fires go.

What has been called the irony of fate is found in the fact that the scene of this appalling disaster was the newest of Chicago's theaters, a building of fireproof construction that justified the name so far as the building itself was concerned—a theater that structurally, perhaps, had no superior in this country or in the world. Little except scenery, decorations and upholstery was damaged by the fierce fire.

It is true that there had been shameful neglect in important details of fitting up, that fire hose on the stage had been delayed, and that fire pails and soda-water fire-extinguishers were absent, and that the ventilating skylights over the stage were blocked so they could not slide open, and that exits were poorly marked; but I have come to believe that had these all been in the condition commonly found in American theaters, the result of this fire might have still been appalling, and it is because I am sure the great lessons of this and the other great theater catastrophes have not been properly heeded that I speak on this topic to-night.

The great lesson of the Iroquois centers around the sudden outbreak, the rapid progress of the fire over the stage, and the fact that most of the deaths occurred within five minutes of the first flame; that death came to nearly all of those who had seats in the gallery, while nearly all of those on the floor escaped.

The great lesson of the Iroquois fire was only a repetition of a lesson that has been given several times before and each time forgotten.

The recurring formula is:

(1) A stage crowded with scenery.
(2) The sudden spread of the flames over this scenery.
(3) The opening of a door in the rear of the stage, an inrush of air.
(4) Scant smoke vents over the stage, an outburst of smoke under the proscenium arch.
(5) Death to those in the galleries.

In 1881, at the Ring Theater disaster in Vienna, with about 1800 in the audience, careless lighting ignited a "hanging border;"

a large door in the rear of stage was opened, letting in a blast of air that drove the smoke through the proscenium arch; the iron curtain could not be lowered; special exit doors were found locked; 450 were killed, *mostly in the upper gallery.*

In 1887, at Exeter, England, fire caught on a stage crowded with scenery. *Within about five minutes* from the outbreak of the fire, 200 were killed, *mostly in the upper gallery.*

In 1876, at Conway's Theater, Brooklyn, N. Y., the stage was crowded with scenery; a border caught fire; the blast of suffocating smoke was increased by the opening of large doors in the rear of stage; about 300 were killed, *all in the upper gallery.*

Note the suddenness, the suffocation, and that *the fatalities are nearly all in the galleries* and that these old descriptions will each tell the story of the Iroquois.

In 1903, at the Iroquois Theater, Chicago, the stage was crowded with scenery. A piece of hanging scenery was set on fire by an electric light. A door at the rear of stage was opened, increasing the blast of suffocating smoke sent into the auditorium. Within from five to ten minutes about 580 were killed, mostly in the upper gallery. Substantially, all of those who had seats on the floor got out alive. Out of about 900 who were in the gallery and the balcony only about 300 got out alive.

The obvious suggestion might be, make the scenery incombustible, and the popular belief taken in from old text-books is that this is a simple matter. Of its difficulties and uncertainties, we will speak later. Suffice it for the moment to say that notwithstanding the paternal care with which the government in England and on the Continent looks after all matters of public safety, and notwithstanding the many recipes in French and English publications for making fabrics incombustible, none of these foreign governments, so far as I can learn, specify that scenery shall be subjected to any process of flame-proofing.

We may make scenery less easily inflammable, so that a match or an electric spark will not ignite the canvas or gauze, but *the efficient fire-proofing of scenery, so that it will not all burn up if a fire once gets well started on the stage, is simply impracticable.* Of all this we will speak later.

The scenery which burned so rapidly at the Iroquois was all made in England, and was first used under the supervision of English law, in the Drury Lane Theater, in London.

We will in briefest manner possible discuss a few of the chief features of the theater risk and means for meeting them.

The Fuel.

The amount of combustible material on the stage in a great spectacular piece is surprisingly large. On the Iroquois stage at the time of the fire there was more than ten thousand square yards of canvas, or two and one-half acres, and in addition about

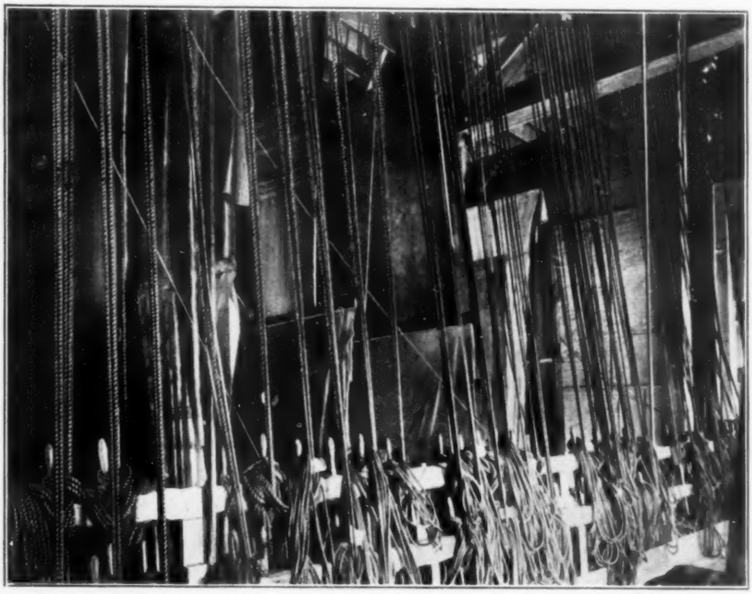

FIG. 1.—A TYPICAL VIEW OVER A THEATER STAGE, ABOVE THE LEVEL OF THE PROSCENIUM ARCH, SHOWING THE CANVAS SCENERY, THE ROPES BY WHICH IT IS RAISED AND LOWERED, AND THE "PIN-RAIL" ON WHICH THESE ROPES ARE FASTENED.

three thousand square yards, or half an acre, of gauze. To hang this required nearly eleven miles in length of ⅝-inch manila rope, and in the frames, battens, braces, profiles and set pieces, the stage carpenter of the Iroquois tells me, after making careful estimate, that there was about eight thousand square feet of white pine lumber. The total weight of this fuel was more than ten tons, all dry as tinder, and all set or hung in a way to give the quickest possible exposure and spread to the flames.

Figs. 1 and 2 will give some idea of how this is hung.

The paints used by the scene painter are not dangerous. They are almost entirely mineral substances put on with water and glue, and they tend to make the fabric a little less readily combustible.

It is very rare that so much scenery is found upon a stage; but if, as is more common, it were only one-fourth part as much as at the Iroquois, it is plain that the fuel supply is sufficient to send out an enormous volume of suffocating gas. Indeed, I have computed that merely the quick burning of this one hundred and sixty pounds of gauze that hung over the Iroquois stage would heat a volume of air equal to that contained in the large space of the hanging loft above the level of the proscenium arch to 1,000 degrees Fahrenheit.

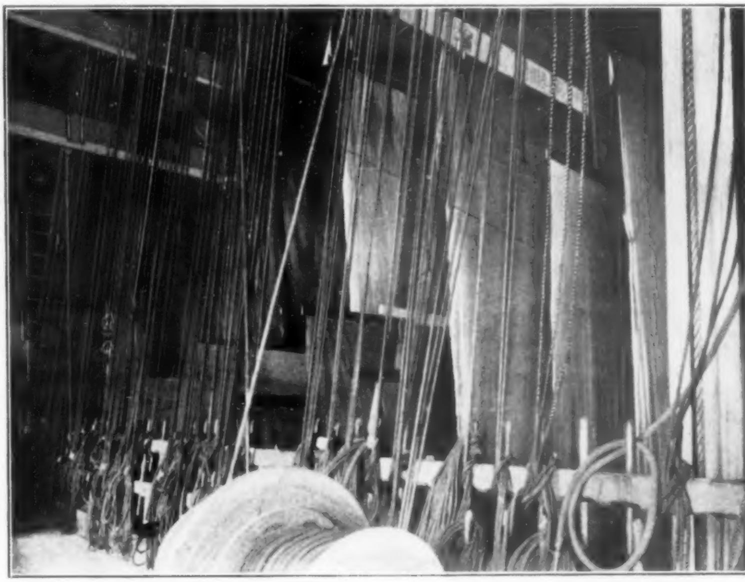

FIG. 2.—ANOTHER TYPICAL "HANGING-LOFT" OVER A THEATER STAGE. NOTE HOW THE NECESSARY ARRANGEMENT OF THE SHEETS OF CANVAS FAVORS RAPID BURNING.

There is good testimony to the effect that in the Iroquois fire only about two minutes' time elapsed after the first spark until all the upper scenery was in flames. Only from three to four minutes' time elapsed before the large space of the hanging loft was so filled with fire that the flames and smoke rolled out be-

neath the proscenium arch into the top of the auditorium; inside of five minutes from the first spark came suffocation and death.

The foremost problem of safeguarding life in theaters is to give prompt and certain vent to this smoke and suffocating gas elsewhere than through the proscenium arch.

CONCERNING THE SMOKE VENTS.

The ordinary construction, with a high spacious chamber for the hanging loft above the level of the proscenium arch, is such

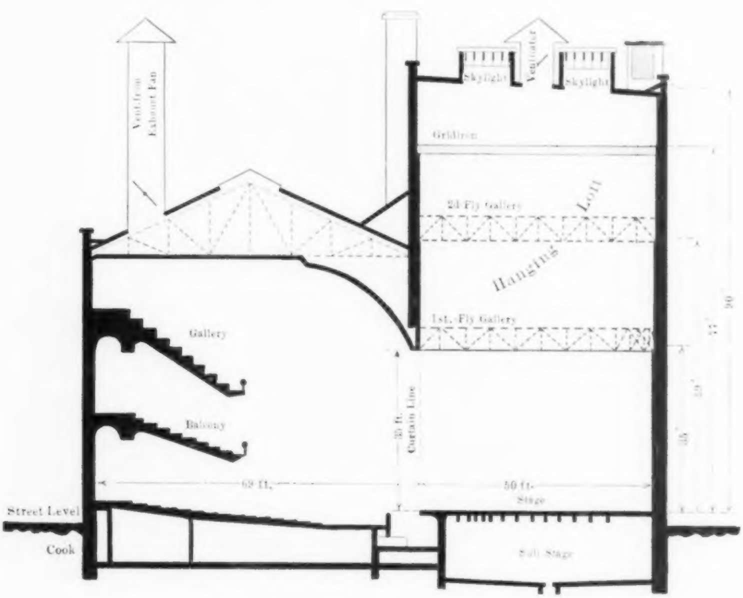

FIG. 3.

that *it is a simple matter structurally to keep this fire and smoke out of the auditorium*, and no matter how great the mass of flaming scenery, a smoke vent of one-eighth or one-tenth the area of the stage, if instantly opened, would probably have saved all of this terrible suffocation at Chicago, at Exeter, at Brooklyn and at Vienna. *This remedy is so simple, so sure and so cheap that it is a crime not to apply it.*

A thoroughly good automatic smoke vent will do more for the

safety of the public than all of the remaining provisions of the most elaborate building law.

As yet, not one theater in ten has it!

Fig. 3 shows a cross section of the Iroquois through auditorium and stage. The form is typical and about the same in all first-class theaters. To one who has not been behind the scenes and climbed up to the gridiron, the surprising thing is the great head room, commonly seventy feet from stage to gridiron and eighty or sometimes ninety feet from floor to roof, and necessarily more than double the height of the proscenium arch, into which are hoisted the great sheets of canvas on which the scenes are painted.

The conditions are plainly similar to that of the fireplace in our living room, magnified ten or twenty diameters. Note how admirably the high space over the stage, screened by the arch, is adapted to give the best of chimney draft, and not give us a smoky fireplace. The roaring fire on our hearth sends ninety or ninety-five per cent. of its heat up the chimney and gives out no smoke into the room, if only the chimney be properly designed and the damper open. An ordinary rule is to make the throat of the chimney at least one-tenth the area of the fireplace opening, or it may be stated that the space through the damper should be one-eighth the area of the hearth, and *when we simply provide an adequate chimney area and a damper that will surely open, we shall have adopted a safeguard that would have saved four-fifths of those who perished at the Iroquois*, regardless of defective curtain, defective exits and absence of fire hose on the stage.

In a way, it has been long recognized there should be a large ventilator over the stage, and one city has copied from another the building law that in the case of New York City reads as follows:

> "There shall be provided over the stage metal **skylights** of a combined area of at least **one-eighth** the area of the stage, fitted with **sliding sash** and glazed with double thick sheet glass . . . the whole of which skylight shall be so constructed as to **open instantly** on the cutting or burning of a **hempen cord** . . . Immediately underneath the glass of said skylight there shall be **wire netting**. . . ." etc.

The evident purpose of the thin glass is to cover the vent with something that will break out under heat if the mechanism for sliding the cover off fails. The wire netting is to catch any piece of broken glass from falling to the stage.

The building law of the London County Council reads much the same, save that its ratio is one-tenth, and perhaps that ordinance is where the rule began.

Some of the leading American cities make the proportion one-tenth. In the revised Chicago ordinance, notwithstanding their fearful lesson, they are content with ventilators of one-twentieth the net area of the stage, because, as one of the Aldermanic Committee gravely assured me, "If the area was made too large, it might cause a down draft."—What stupidity!

The idea of a large ventilator expressed in these rules is all right, but the execution is commonly all wrong, and needs some good engineering to provide a design of damper with careful details that will be sure to work. Note the antiquated, good-for-nothing suggestion of the "burning of a hempen cord," when fusible links have been used on the fire doors in your factories for twenty years! There is no good reason to expect the hempen cord in this position in smoky atmosphere from which oxygen had been largely removed would burn off until after a majority of the people in the gallery had been suffocated.

And in one of the newest and best of the New York theaters I found the ventilator had a broad sheet of heavy canvas laced tightly across its opening with marline, because, as the stage carpenter told me, "the cracks around the ventilator let in too much cold air." No building inspector had objected, and the carpenter could not be made to see any danger. "It would burn off in any bad fire," he said. So it might, but not until the people in the gallery were mostly dead.

The requirement of thin glass in the building law is well meant, but it would be too slow in breaking out. Remember how quickly unconsciousness of suffocation comes in an atmosphere of smoke. The wire netting called for *is a positive danger*, as often applied.

One of the most experienced theater managers in America told me frankly that he knew the smoke vents on the theater which he then occupied would probably not open in winter unless a man should first pry them loose with an iron bar; but, said he, "I have not heretofore seen anything better," and so after the Iroquois he had set his stage carpenter at work to invent something.

Doubtless, there are good smoke vents here and there that have been designed and built with skill and conscience, for the problem is not so very difficult, but I have not yet seen one of these vitally important pieces of apparatus in which the design

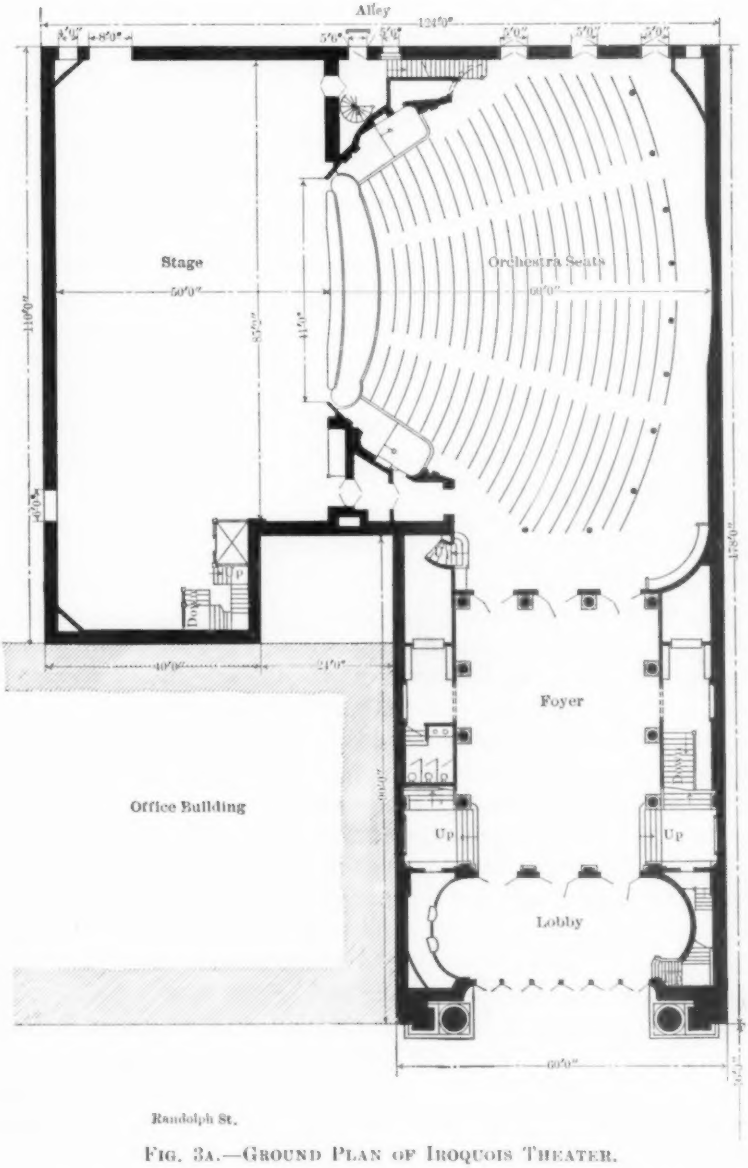

FIG. 3A.—GROUND PLAN OF IROQUOIS THEATER.

had been worked out with reasonable degree of perfection of detail. "Something good enough to pass the building inspector" appears to have been the current specification, instead of the

proper specification of "*a vent of ample area that will be sure to open wide, instantly, without human intervention, and that cannot be stopped by warping, settlement, obstruction, frost, snow, rust, dirt, or ordinary neglect.*"

I do not know who first fixed this ratio of one-tenth for size of ventilator, the same figure that appears in the old rule of the London County Council. Its author may have built wiser than he knew, or may have taken it from the well-proved ratio of the common home fireplace chimney. It works out as safe when computed mathematically on theoretic grounds from the uncertain data. The material is so favorably disposed for ignition that the rapidity of combustion is largely a question of the air supply.

I am led by computation and precedent and the need of some factor of safety, to concur in the wisdom of the ratio of one-eighth or one-tenth, as already specified by the building laws of the great majority of our American cities, and I believe it wise to base it upon the gross area of the stage floor rather than upon proscenium opening or cubical contents of the stage.

I have seen here in New York, in a recent theater, a case where the inspector had, perhaps temporarily, forgotten the wording of the law and figured it on the area directly in behind the curtain, omitting much of the floor space at the sides. This is wrong because, given a large stage, there is a well-proved tendency to permit an unnecessarily large amount of combustible material upon it, and it not infrequently happens that the scenery of next week's new attraction may be found stored at the side and rear during the Saturday night performance.

I was earnestly desirous of making some practical tests in the Iroquois, for the additional flames could have caused no injury beyond that already wrought, and the practical demonstration which I am confident would have been given regarding the efficiency of automatic sprinklers and proper smoke vents would have hastened their general introduction. Of course such tests would have been made under greatest precautions and with city fire department at hand in force, but while many of the most important interests readily consented a few of those owning surrounding property objected.

The Austrian Experiments of 1885.

Following the great theater fire in Vienna, a committee of the Austrian Society of Engineers (Vereines der Techniker, in Ober

Osterreich) built a model of the Ring Theatre on one-tenth of its lineal scale, which thus contained only one-one-thousandth of the cubic contents of the original, and made many tests and experiments.

The experiments were divided into two groups, the first comprising those in which no ventilators were opened over the stage, while in each of the experiments of the second group two ventilators were opened, having a combined area which according to the scale of their drawing I find was very nearly one-tenth of the area of the stage. In the first series of tests made by igniting sheets of paper hung to represent the scenery, but containing proportionally far less combustible material than is often hung on a theater stage, they found that the expansion of the air caused by the heat *quickly forced the curtain outward from the proscenium arch*, and within about twenty seconds from lighting the fire, this heating of the air produced an excess of atmospheric pressure, much greater than that of the ordinary pressure of city gas, thereby explaining why it was that the lights in the Ring Theater became so quickly extinguished after the outburst of the fire.

In the second series of these Austrian experiments, their models of the ventilating shafts were closed by sheets of paper, and as soon as these smoke vents burned open, all excess of air pressure disappeared from the auditorium, and indeed, the updraft drew the proscenium curtain inward over the stage.

During these experiments an unexpected warning was given against covering smoke vents by wire screens, for it was found the flying bits of charred paper carried up by the draft almost completely closed them. To show how little this warning of the Austrian Society of Engineers has become incorporated in current practice, I may call attention to the building law of New York City, which *requires* that underneath all of these skylight openings designed as smoke vents, wire netting must be stretched; the law apparently never considering how quickly this will become so clogged as to destroy in large part the utility of the smoke vent. At my visit to the remodeled Iroquois, I found the openings in their new ventilating shafts screened by wire netting in a way that would probably *within a minute's time put them into a condition of uselessness* because of the fragments of burning cloth and embers with which they would be immediately covered under the strong updraft, all of course with approval of architect and building inspector!

The committee of the Austrian Society of Engineers concluded that the outburst of flame and smoke into the upper part of the auditorium and the extinguishment of gas lights in a theater *could all be prevented by providing adequate smoke vents* over the stage, and places these smoke vents as the feature of first import in safeguarding life in theaters, and says that without them emergency exits and fire curtains will be of no avail; and in this conclusion I most heartily concur, for I had independently reached it from my investigations following the Iroquois disaster, prior to learning of the experiments of the Austrian engineers.

Regarding the mechanical construction of these smoke vents, the Austrian committee says, "It is necessary that these be opened instantly upon the outbreak of the fire; mechanical contrivances of iron to be operated by human means will certainly fail, for, according to all experiences in theater fires thus far, fright on the part of the employees prevents the use of such arrangements." They warned against automatic contrivances whose action may be interfered with in consequence of rust or expansion by heat, and against sheet iron valves falling inward by their own weight, which might be restrained from falling open by the excess of pressure due to updraft, and finally recommended that these shafts be closed by a quickly combustible tissue of hemp or jute covered with varnish or celluloid, and with a hole about one and one-half inches in diameter in the center to invite quicker ignition. Our Austrian friends were unfamiliar with the American fusible-solder link, which is certainly quicker and safer and in every way far more practical than any such tissue of varnished hemp.*

**Austrian Experiments of* 1905.

While revising the proofs of the above for publication in the yearly volume, report comes of a second series of tests on a somewhat larger scale, made on and about Nov. 22, 1905, in Vienna at the expense of the Austrian government, on recommendation of the Austrian Engineers and Architects Association. From the brief preliminary report in Eng. News of Jan. 18, 1906, the following is taken:

The model theater, constructed of reinforced concrete especially for these tests, had a stage 24.6 ft. wide, 19.7 ft. deep, 25.3 ft. high, with a proscenium opening 11 ft. wide and 8.5 ft. high. The auditorium was 18 ft. wide, 23 ft. deep, 15.4 ft. high, or in general had about $\frac{1}{3}$ the linear dimensions of the ordinary theater, and therefore about $\frac{1}{27}$ of its cubic capacity.

The tests made by burning old scenery and sheets of paper, representing proportionally the amount of combustible for two performances, showed that with smoke vents of total area of 11 per cent. of the stage area opened, the smoke ascended through these vents over the stage with no suggestion of danger to the persons in the auditorium, except that near the proscenium opening the heat was somewhat severe.

Fusible Links for Opening Smoke Vents.

These links have been in common use on automatic fire shutters and fire doors in our factories for twenty years. Three types of these links are shown in Fig. 4. Each is reliable,

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On the other hand, in tests with stage vents closed and curtain down, it was bulged out toward the audience and lifted from the floor at the bottom, and the auditorium was soon filled with smoke.

In a later experiment with sprinklers spraying the fire, on opening a door or ventilator in the auditorium gallery some steam and hot gases were drawn into the auditorium, notwithstanding the stage smoke vents were open.

As a whole these tests again demonstrated the importance and the remarkable efficiency of a smoke vent over the stage, of about 11 per cent. of its area.

It is of interest to note that these two sets of Austrian experiments have given a complete answer to two of the puzzling questions of the fire at the Ring Theatre. At the inquest the man was sought who was supposed to have turned off the gas from fear of an explosion, thus leaving the house in darkness while the audience and actors were struggling to escape; he was not found. Both series of these experiments on the theater models show that a back pressure of air in the auditorium more than sufficient to force the gas back in the pipes, and thus extinguish the lights, was produced by the rapid expansion of the air over the stage due to the heat of the fire. Indeed, this quick back pressure was found sufficient to account for the bursting open of the large scene door at the back, which it had been supposed was opened inadvisedly, thereby causing the draft which blew the suffocating smoke into the auditorium.

I have not the full report of these later tests at hand. In studying the reports of the Austrian tests of 1885, I am unable to believe that the back pressures due to expansion of air are ever likely to be so large in an actual theater fire as those developed in the model tests and carefully measured on the manometers. I saw no evidence of so great pressures at the Iroquois, and failed to find evidence in the testimony of the eye-witnesses, although the conditions were favorable for very rapid burning. I have no doubt there may have been sufficient pressure momentarily, at the end of the first half-minute or full minute, to blow the curtain strongly outward, but the absence of scorching of wood and textiles around the opened rear stage door shows conclusively that after this was opened the air current there was continuously inward.

In the Austrian experiments of 1885, with smoke vents closed, air pressures were developed momentarily at from 20 to 30 seconds after lighting the fire—as great as ¼ lb. per square inch, or equivalent to 5 or 7 inches of water column or 32 to 38 lbs. per square foot! With smoke vents covered by thin paper which quickly burned open the excess of air pressure on the stage was only momentarily equivalent to 0.07 inch of water column, with no excess observed in the auditorium.

The preliminary reports of the Austrian experiments of 1905 show that with smoke vents closed, even a steel proscenium curtain was no sufficient safeguard for the audience. The air pressure due to expansion held it from lowering promptly, and when lowered the suffocating gas and flames were driven past its loosely fitting edges into the auditorium. With smoke vents open a proscenium curtain was hardly necessary.

practical and successful, as proved by years of use. They can be obtained from any of the manufacturers of automatic sprinklers.

It is strange almost beyond belief how slowly and scantily these have found their way into the fire protection of theaters.

These links melt open at about one hundred and sixty-two de-

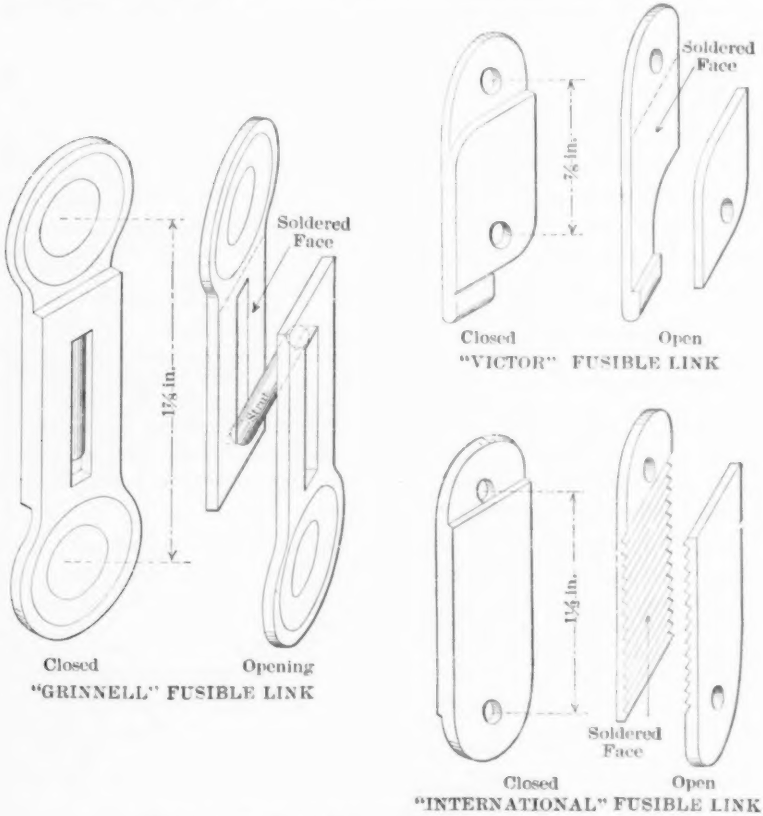


FIG. 4.—FUSIBLE LINKS FOR FIRE PROTECTION.
SCALE, FULL SIZE.

grees Fahrenheit, and thus will open long before flame reaches them. Their cost is trifling. They are stamped out of sheet brass, soldered with "fusible solder," the formula for the 162° F. alloy being

Tin	12	per cent.
Lead	25	" "
Bismuth	50	" "
Cadmium	13	" "
Total	100	" "

Links like those shown in Fig. 4, tested to immediate rupture, will break under a load of about two hundred to five hundred pounds, but can be trusted to sustain continuously a load of only about fifty to one hundred pounds. Our tests show that the solder becomes somewhat weaker in warm air than in cool air. Probably in an atmosphere of 100° F. the links would safely sustain 10 per cent. to 20 per cent. less load than in an atmosphere at 60° to 70° F.

All of the known solders that fuse at low temperature are subject to stretching or "cold flow" under long-continued loads, unless these loads are made extremely small, and one of the most important features in the design of any such link is to make the direct stress upon the solder small and in tension over a large area, rather than by shear.

The links shown in Fig. 4 will open with about the same promptness as an automatic sprinkler. In a test oven, immersed in an atmosphere of four hundred degrees Fahrenheit, these various forms of link open in 35 to 100 seconds; in five hundred degrees, in 25 to 75 seconds.

At top of rigging loft over a fire like that on the Iroquois stage, fusible links probably would open within 30 to 60 seconds after the blaze got a good start, and in ample time before the smoke would burst out under the proscenium arch. There is ample space in the hanging loft to pocket the smoke for the first minute or two. At the Iroquois fire statements of eyewitnesses indicate that it was probably fully two minutes before the smoke rolled out into the auditorium. I was surprised to find on measurement of the Iroquois plans that the volume of air space above the level of the arch was greater over the stage than over the auditorium.

The sensitiveness of these links or their quickness of action under moderate degrees of heat depends on the thinness of the mass of metal to be warmed up, and therefore on the rapidity with which it absorbs heat enough to melt the solder. These two characteristics—the weakness of the fusible solder under long-continued strain, and the necessity for rapid absorption of heat, limit the size and strength of fusible link that can be employed; but it is easy to so design the connections that the strain will be about fifty pounds, thus large enough to override dust, rust and petty derangements and small enough to be within the capacity of the fusible link. In many situations a link is desired of such form

and size that when inserted in a rope it can run over the ordinary pulley.

Imperfections of Smoke-Vent Design.

Concerning the design of smoke vents, those that I have seen in actual use have been, with hardly an exception, imperfect pieces of mechanical design. At certain of the most recent New York theaters I have found the type which appears to be the favorite for meeting the New York building law, set with such a clearance as to give a very unnecessary degree of ventilation, which tempts the theater mechanic to stop the draft by some means that may prove dangerous. It is, moreover, so heavy and unwieldy that it cannot be frequently tested by opening and closing, and to wait for the burning of a hempen cord to open a device of this kind should be regarded as criminal negligence when it can be done so much better and quicker by the automatic fusible link.

Smoke-Vent Designs by the Author.

To meet the proper suggestion that one should not merely criticise without presenting a better device, and as a means of illustrating that the problem can be solved along various lines of design, I have worked out two models, shown in the accompanying drawings. I am certain that with the experience to be gained in constructing one after another, these designs could be improved upon. It is desirable that the total smoke-vent area of one-eighth or one-tenth the stage be subdivided into four independent units for convenience in size and for the further safeguard that should one stick there are three others left.

The fundamental requirements for a theatre smoke vent are:

1st. **Absolute certainty** of opening by force of gravity, in spite of neglect, rust, dirt, frost, snow, or expansion by heat, twisting or warping of the framework.

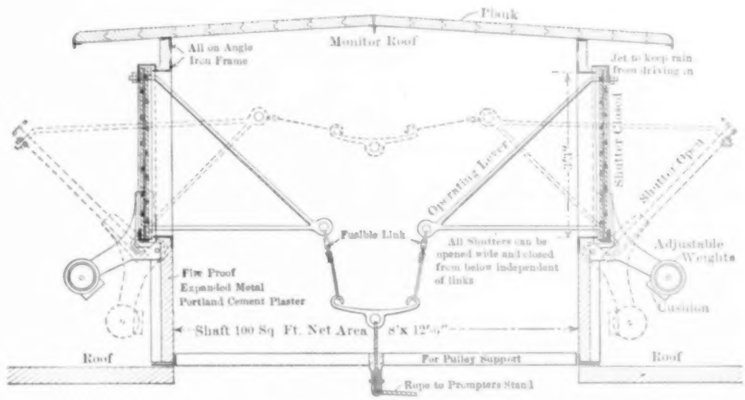
2d. **Quickness** of opening to be secured by automatic links of the thinnest metal practicable, and also by controlling the doors by a cord run down to the prompter's stand and to the station of the stage fire-guard.

3d. **Simplicity** and **massiveness** of the operative mechanism of the smoke vent. This should be designed not as a watchmaker would build it, but more according to the standards of railroad service or rolling-mill practice. The counterpoise weights should be heavy, and a constant tension on the re-

lease cord of upward of thirty or forty pounds so that rust, cobwebs or temperature changes may not be of noticeable effect in the resistance to be overcome.

4th. **Daily Tests.**—It should be of such form that it can be tested daily, or at least at the weekly inspection, by partially opening it, preferably closing it again by means of the cord running to the prompter's stand. It may perhaps add to the safety if it is of such design that it can be used whenever needed for the ordinary ventilation of the stage, summer or winter, rain or shine, thereby keeping it under constant view and bringing into immediate notice any difficulty about its opening or lack of repair.

In the first of these designs submitted, shown in Figs. 5, 5a, 5b,

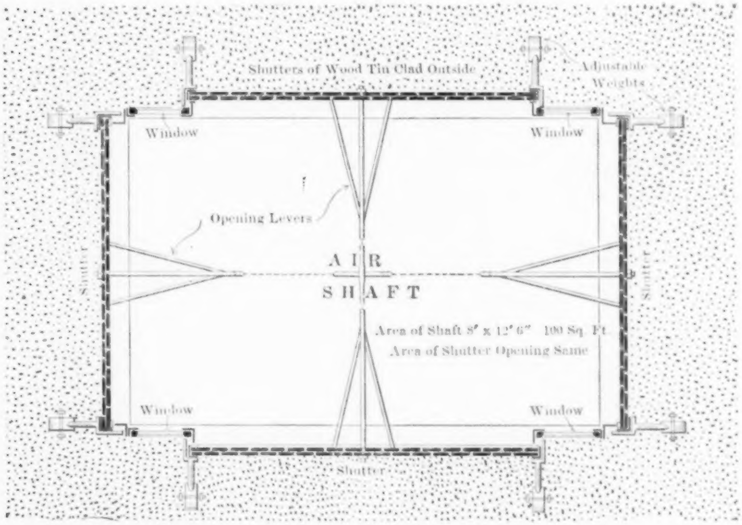


SMOKE VENT. MONITOR TYPE
VERTICAL SECTION

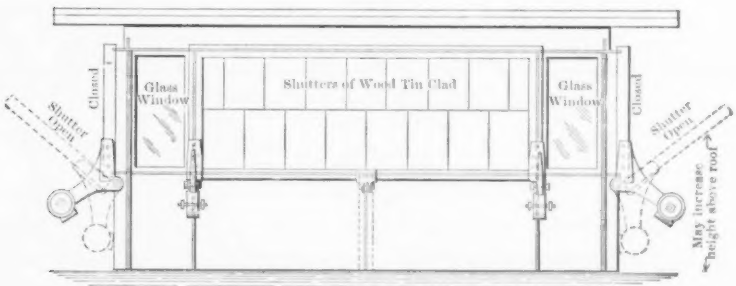
FIG. 5.

5c, 5d, the opening, eight by twelve and one-half feet, of which four would be needed over the stage of ordinary size, has a roof for protection from rain and has vertical sides that contain four small windows for admitting abundant daylight to the rigging loft, but which can be closed by ordinary window shades for dark scenes. All necessity for the wire screen is avoided. The vertical shaft may well be three to five feet taller than shown. The four shutters fall *outward* lest the pressure of the updraft tend to hold them shut, and are pulled open *by force of gravity*, opening to the full area called for. The pull on the rope holds them

against their seat, which, if made with a thin edge pressing loosely against fibrous material, as shown, will be more tight against cold-air drafts than a common window sash or house door. Fusible links are inserted in each of the four branches of the



SMOKE VENT. MONITOR TYPE
HORIZONTAL SECTION
FIG. 5A.



SMOKE VENT. MONITOR TYPE
SIDE ELEVATION
FIG. 5B.

cord. *No sprinkler should be placed up within the monitor containing these links*, lest perchance the sprinkler open first and chill the links, and care should be taken that the links are of a thin, quickly sensitive type.

In the second design, Fig. 6, the sliding type is used. This obviously cannot be used as an ordinary ventilator in rainy days.

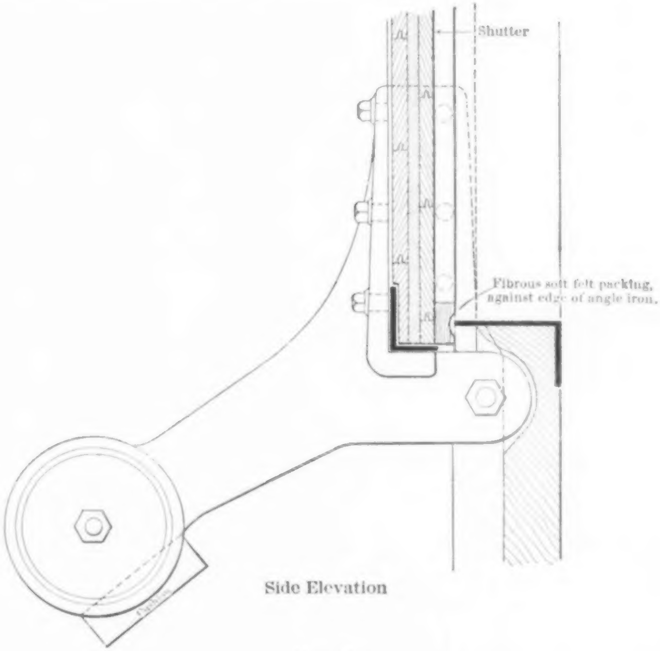
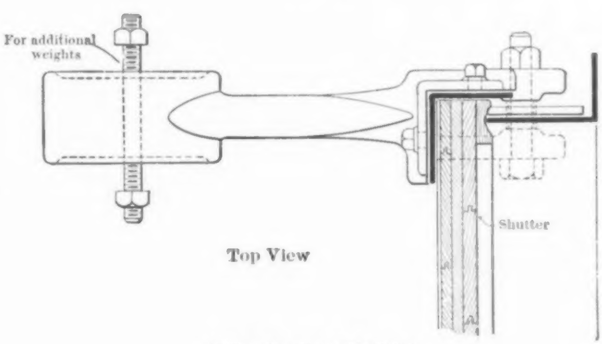
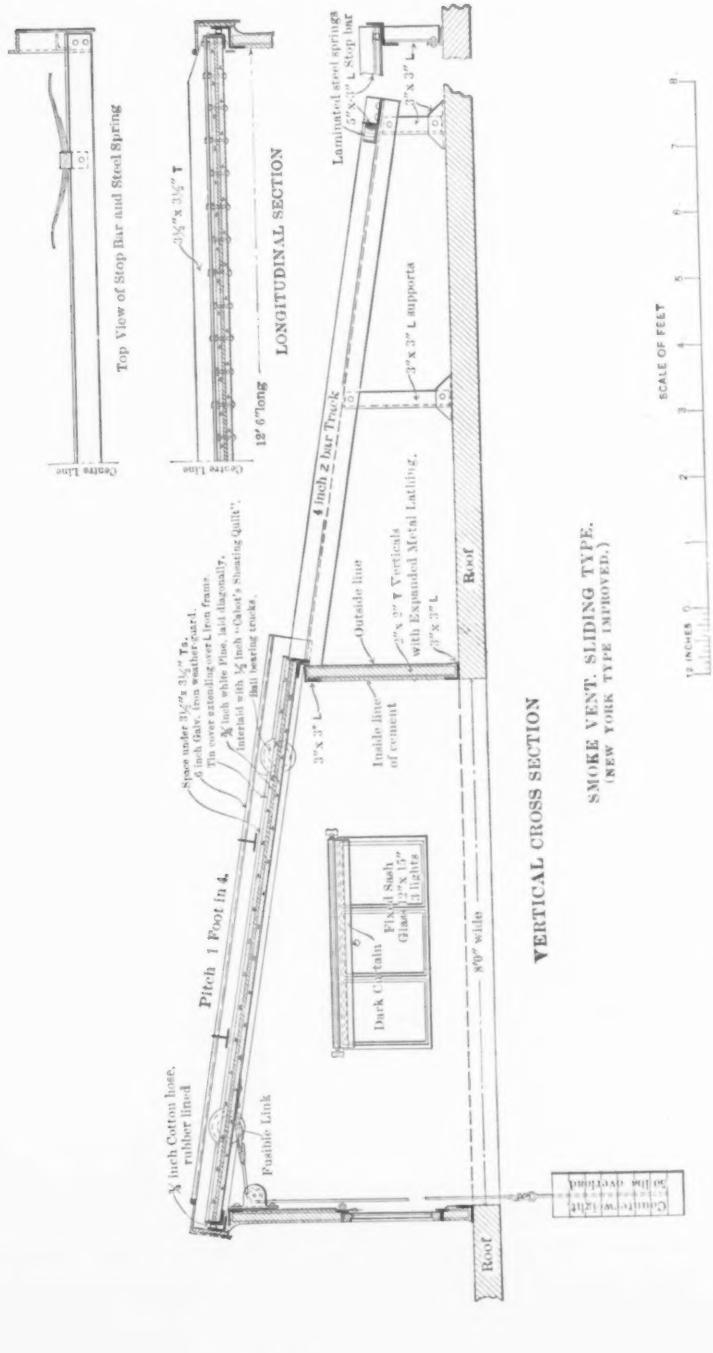


FIG. 5C.



DETAIL OF WEIGHT
FOR SHUTTER OF SMOKE VENT, MONITOR TYPE
FIG. 5D.

The special effort in remodeling this from the current New York type has been, first, to place the glass in the vertical side so that no necessity remains for a wire screen to catch any broken glass.



SMOKE VENT. SLIDING TYPE.
(NEW YORK TYPE IMPROVED.)

FIG. 6.

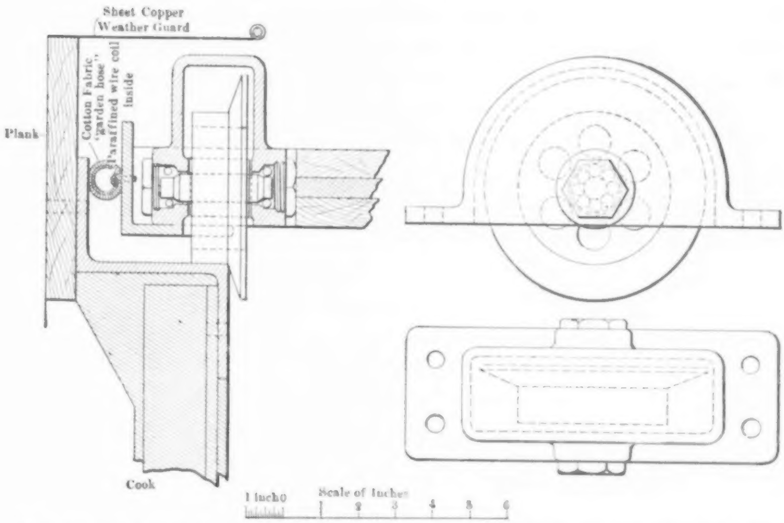


FIG. 6A.—DETAIL AUTOMATIC SMOKE VENT. SLIDING TYPE. DETAILS OF ANTI-FRICTION TRUCK AND OF COLD-AIR STOP.

Second, to provide a better track and trucks and arrange the joints so that the leakage of air through the clearance space would not tempt the janitor to close the space by something that may interfere with the sliding open. Common cotton fabric garden hose, paraffined outside and with an elastic lining or with a thin wire spiral inserted, meets the need for a yielding, non-adhesive packing, if applied as shown.

A third sketch, Fig. 7, shows an arrangement of a safety ventilating shutter that sometimes can be conveniently placed in the brick wall near the top of the rigging loft.

It is dangerous economy to be niggardly about providing the best design and workmanship for the smoke vents, because these are the most important of all the structural safeguards to life in a theater.

AUTOMATIC SPRINKLERS.

The second safeguard in order of importance is, in my opinion, complete equipment with automatic sprinklers over the stage and throughout all rooms and nooks and corners except in the auditorium.

It is now twenty-five years since a former vice-president of this Society, the late Col. Thos. J. Borden, of Fall River, intro-

duced automatic sprinklers in two Fall River cotton mills under his charge, throughout picker rooms, card rooms and spinning rooms. These were put in by Mr. Frederick Grinnell, lately deceased, also a member of this Society, to whom more than to any other man credit should go for the development of this greatest of safeguards against fire. Since that time the factory insurance companies have been slowly led by their wide and varied experience

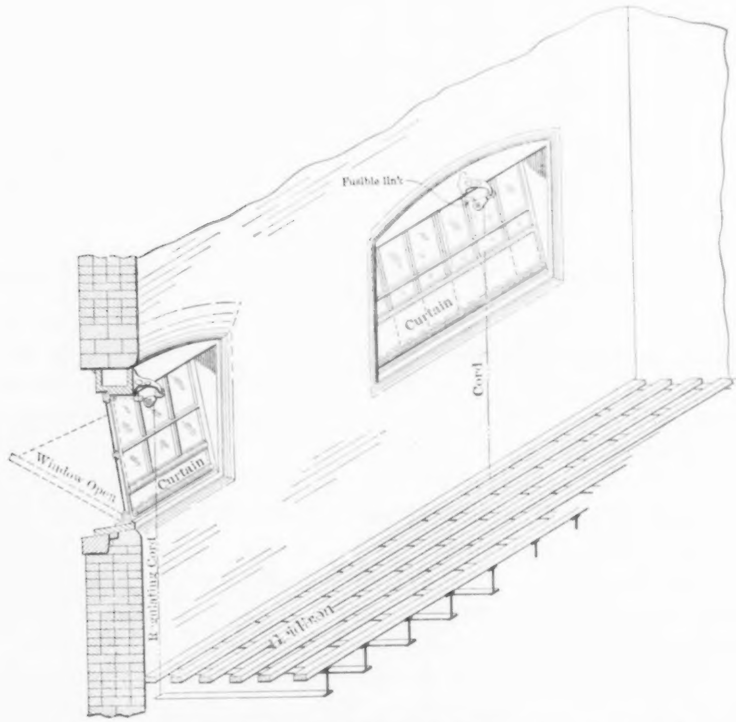


FIG. 7.—AUXILIARY SMOKE VENTS IN WALL ABOVE GRIDIRON.

of 25 years, in thousands of factories, to urge automatic sprinkler protection as the foremost of all safeguards against fire. In the beginning they were recommended cautiously, and only over those portions of factories believed to be the most hazardous. Gradually, as their merits were proved, they were called for throughout wider areas, until it has come to pass that out of about two thousand of the largest factories in America, which entrust their insurance and the supervision of their fire protection to an organ-

zation with which I have long been connected, substantially every ten feet square in mill and storehouse is under automatic-sprinkler protection. We have records of more than a thousand factory fires that have occurred under sprinkler protection, covering a great variety of conditions, and from my own experience in what sprinklers can do to control a fire under adverse circumstances, I unhesitatingly recommend them as the best of all known means for promptly controlling a fire that has once got good hold in the scenery upon the stage of a theater. Like everything else in the mechanical line, they should be skillfully put in.

It has been claimed, as a matter of intuition, by some who have not closely followed the experience with sprinklers over fires, that under the high rigging loft of a theater, sprinklers at a distance of sixty or perhaps eighty feet above the floor of the stage would be so remote from the flames that they would not open with sufficient promptness to be of material service. I am confident that this is untrue. The hot air from a fire quickly travels over a vertical distance of sixty or eighty feet. Not more than five to ten seconds' time would be required for this rise, and the conditions for pocketing and confining the heat to a small area in the top of the rigging loft of a theater are much more favorable than in many portions of factories where sprinklers are found to work successfully.

The rainfall from a series of automatic sprinklers carries tenfold more water than that from densest ten minutes of the heaviest thunder shower of the ordinary year.

With eighty square feet to the sprinkler and the ordinary water pressure, this sudden artificial rainfall would be at the rate of about twenty-five inches per hour.

One series of sprinkler heads should be placed below the gridiron, and preferably another series above it, these not being vertically over one another. Those in the top series are as likely to open first, but it is well to be liberal and provide both series. A line should also run along the lower outer edge of each fly gallery. With care, a skillful sprinkler fitter can readily place and guard all the heads and pipes so the danger of breakage need be no greater than in a factory. The one hundred and sixty-two-degree solder should be used and the piping should be on the "wet system."*

* The foundation patents on these devices have expired. Half a dozen different good makes are on the market. Under the somewhat difficult conditions of

Stage scenery, while exposed to very rapid ignition, is equally well exposed to very rapid drenching, and the fact that we have so few actual records of what sprinklers can do in controlling a fire on the stage is due to the few instances where sprinklers have been installed in theaters, or have had an opportunity to demonstrate the work of which they are capable. At least there have been no failures, and there are several most notable successes to their credit. The first was in a case where they had been put into a theater because Mr. Cumnock, a factory manager who was one of the stockholders, had been satisfied of their efficiency by fires that they had extinguished in his cotton mill.

This was at a theater in Woonsocket, R. I., in which a gauze piece took fire from the border lights prior to the performance, and sprinklers opened under the gridiron sixty-five feet above the floor, while other sprinklers opened under the roof eighty feet from the floor.* At theaters in Philadelphia, in New York City and in Providence, R. I., there have been notable instances of fires when the audience was absent, from spontaneous combustion and overturned lamps, in which the sprinkler extinguished the flames, and from Manchester, England, a case is reported of a fire in a "gauze sky," between the acts, extinguished by four

installing them about a theater the cost for sprinklers, pipes, fittings and erection will average, perhaps, $5.00 per head. In factories the cost averages about $3.00 per head.

The heads are placed from 8 to 10 feet apart, and fed from the public water supply or from elevated tanks. The cost of tanks and main water supply is not included in the estimate above, and is liable to bring the total cost to $10.00 per head in theaters, varying widely according to circumstances. In cities with low pressure a large tank for water supply adds materially to the expense.

With a stage of medium size, say 40 x 60, or 2,400 square feet and 80 square feet per sprinkler, 30 heads would be required on the ceiling above the gridiron, and 30 more just below the gridiron. About 10 heads more would be needed in lines along the edge of fly galleries, and perhaps 30 more beneath the stage, making, say, 100 sprinkler heads for the stage portion of a medium-sized theater.

The dressing-rooms, auditorium, basement, carpenter's room and various odd corners are likely to call for another hundred sprinklers, so that 200 sprinklers would perhaps be an average equipment.

Some very large theaters have taken 500 sprinkler heads.

The saving in cost of insurance commonly goes far toward paying good interest on this cost, leaving the safeguarding of life as costing little or nothing.

* This case was so complete an answer to the suggestion that sprinklers at the top of a high stage are too far off to work with promptness, that I sent an engineer to measure the distances and sketch the surroundings. These sprinklers worked so effectually that the wooden gridiron was not burned. The sprinklers appear to have put out this fire after it got away from the man with the stage hose.

sprinklers thirty feet above the flies so promptly that although the stage and scenery were wet, the performance went on without the audience knowing just what had been going on while the curtain was down.*

The reasonable safety of the public requires that automatic sprinklers be persistently urged, followed up and demanded under

* Abbreviated from *The Standard*, Boston, February 3, 1906.—At the Colonial Theater, Jan. 20th, the house held a large Saturday matinee audience. Near the close of the performance a strong odor of burning wood was noticed. The automatic alarm and sprinkler signals worked perfectly, and when the firemen arrived it was found that a sprinkler head *had completely extinguished the fire*, which had caught in a box of properties located under the stage. "It should be required by law that all places of amusement be thoroughly equipped with sprinklers and alarms. There are several theaters in Boston not so equipped."

From report of Committee on Surveys, N. Y. Board of Fire Underwriters.—Nov. 29, 1905, Grand Opera House, New York City. About 7 A.M. *fire originated in a quantity of scenery* at side of stage. Cause unknown. Thirty-six automatic sprinklers opened and held fire in check. The sprinkler alarm called watchmen, who summoned Fire Department, which completed extinguishment. That 36 sprinklers were unsoldered proves a lively blaze. (That fire could originate in a lot of scenery is interesting in view of the fact that law requiring flameproofing of all scenery is understood to be now enforced in New York City.)

At the Kensington Theater, Philadelphia, July 30, 1895: Fire occurred at night, probably from spontaneous combustion in a newly painted drop curtain, done in oil colors, and communicated to a side-piece that stood against the proscenium wall. A single automatic sprinkler opened and thoroughly drowned out the fire, which the owner, Mr. John W. Hart, believes would otherwise have quickly spread and wrecked his theater. He writes enthusiastically recommending automatic sprinkler protection for all theaters.

At the Providence Opera House on Sunday morning, September 23, 1900: Combustion started in some garments left on a small trap below the stage—perhaps from a cigarette after the performance of the previous evening. One sprinkler head located about eight feet above opened and the sprinkler alarm summoned the protective department, who found the fire practically extinguished by the water from the sprinkler.

At the Casino Theater, New York, January 11, 1900, about 8.15 P.M.: Fire was started by the upsetting of a lamp in a dressing-room filled with flimsy material. Two automatic sprinklers opened and, it is said, practically extinguished the fire, although the stage fire hose was also brought into service.

At the Queen's Theater, Manchester, England, on July 20, 1899: "As gas was being lighted in the wings, a gauze sky caught fire and ignited several curtains or cloth hangings near it. Soon there was a big blaze, but four sprinklers situated about 30 feet above the flies opened, almost immediately, and before the fire brigade arrived the fire was out."

At White's Opera House, McKeesport, Pa., December 6, 1903: A letter from the proprietor states—We had a slight fire in the basement yesterday and about eighteen sprinklers opened. Thanks to the sprinkler equipment the damage to the building will not exceed $50.00.

At Keith's Chestnut Street Theater, Philadelphia: It is reported that a fire

the building laws and police laws until every theater using movable canvas scenery has this protection over its stage. There are half a dozen well-proved makes, and plenty of experienced sprinkler fitters can be found to properly erect them.

Possibility of Leakage of Automatic Sprinklers.

A leading argument against automatic sprinklers has been the possibility that they would break open when there was no fire, and thus injure the scenery. We have statistics to show how extremely small this danger really is.

Our records, when I last had them compiled, showed that out of a total of something over three million sprinkler heads scattered through more than two thousand different factories, losses from premature discharge were occurring at the rate of about fifty sprinkler heads breaking open per year. This proportion of one sprinkler in each sixty thousand springing a leak per year, when applied to the conditions in a theater that would commonly have less than one hundred and fifty sprinkler heads over the stage, although they were put in both under and over the gridiron and under the fly galleries, would give a probability at any one particular theater of a leak over the stage once in four hundred years.* Should we admit, which is not certain, that the danger of knocking one of these sprinklers open by a blow is greater in the theater than in the factory with its moving machinery, and assume even that they break tenfold more often, it is plain that this danger of leakage is no just ground for excluding sprinklers from over a theater stage.

Our insurance companies do not hesitate to recommend them

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occurred in one of the dressing-rooms, presumably by fabrics coming in contact with gas jet, although this jet was protected by a globe. The sprinkler extinguished the fire without outside assistance, and, in fact, on the arrival of the employees no fire existed, but some dresses were burned and the cleats to which they had hung were charred.

* Later, in going over the replies to my circular of inquiry sent to managers of all known sprinklered theaters in United States and Canada, I find that cases of the accidental bursting open of sprinkler heads have occurred in far greater frequency than is found in factories. Perhaps half of the theaters reporting have had one or more such accidents, due, in nearly all cases, to allowing the temperature to fall so low that one or two sprinkler heads have frozen and burst. In no case does it appear that any serious damage was caused. Obviously, these accidents should be charged to carelessness and not to defects in the sprinkler head; and, obviously, an accident of this kind will seldom be allowed to happen more than once in the same theater.

for a packing and storage room over a quarter million dollars' worth of delicate silks or finest textiles, and so little do we fear the premature discharge that in the fire insurance we guarantee against this water damage in our fire policies with no additional charge. Our careful records show that we are paying for water damage by the premature discharge of sprinklers and the bursting of their pipes and fittings from frost, blows, carelessness and inherent defects, about 5 cents per year per thousand dollars of value covered!

The idea that the fine spray or rain of water from a single opened sprinkler head falling vertically and probably invisible to most of the audience could produce a panic within the audience, however much it might disturb the chorus, is too absurd for serious argument.

Sprinklers, although not so generally used over the stage as they ought to be, have been introduced here and there, and in some cities quite generally. The great theater at Bayreuth, Bavaria, the home of the Wagnerian opera, was completely fitted up with automatic sprinklers eight years ago, 666 sprinkler heads being installed. I now have the record of about one hundred and fifty theaters that have been sprinklered. I sent a circular letter to the managers of many of these theaters asking for their experience. In no case did I receive an adverse criticism, and in the majority of cases they speak in most appreciative terms of the value of this safeguard.

FIRE CURTAIN.

The third of the safeguards demanding investigation is the curtain for closing the proscenium arch.

With good smoke vents and complete automatic sprinkler protection over the stage, and with ample stairways from galleries, it is probable that the audience could escape from a situation as bad as that in the Iroquois, notwithstanding there was a very poor fire curtain, or perhaps no curtain at all; but in theaters, as in factories, it is wise to have a second and even a third line of defence, lest the first happen to be inoperative in the moment of need.

The fire curtain for covering the opening under the proscenium arch in nearly all American theaters outside Chicago, at the present time, is made from a heavy canvas woven from asbestos fiber; and in English theaters the asbestos curtain appears to

have been steadily gaining in favor because of its less weight and smaller cost in comparison with a curtain of sheet iron, stiffened by iron ribs. In Chicago, because the failure of the Iroquois asbestos curtain, and with the excellent corrugated iron curtain of the Auditorium before them, the Aldermanic Committee has made the steel curtain the rule. Chicago to-day leads the country in the substantial quality of its proscenium curtains, and in the present state of the art they merit little criticism except in their lack of a positive down-haul and their need of better holding and guiding in iron channels at the edge.

Like nearly all steel-ribbed shutters, these steel curtains will warp and twist off their seats under ten to fifteen minutes of exposure to a severe fire unless securely held at edges, and should smoke vents be closed and sprinklers lacking and a back door open, their loose fit would let volumes of suffocating smoke and tongues of flame pass by their edges into the auditorium. With the smoke vents open and the draft therefore inward, they will serve their purpose until the audience has escaped and the firemen have arrived.

Special attention was directed to the asbestos curtain in the Iroquois fire from the fact that the curtain, *although promptly let loose, failed to fall;* because, as some say, it was blown outward from the stage by the strong current of air; or because, as others say, it caught on certain of the electric light shields.

It is a fact that the asbestos canvas soon fell as mere rubbish to the stage, but so little that resembled a piece of asbestos canvas could be found in the wreckage on the stage that it was for some time believed that the curtain had not been made of asbestos.

It is now certain beyond question that this Iroquois curtain actually was made of a good, ordinary commercial quality of asbestos canvas, but somewhat thinner than the very best, and it is doubtless true that this Iroquois curtain was just as good as those which hang to-day in the great majority of our theaters. I personally found fragments of this asbestos cloth in my first examination of the stage while everything was just as the fire left it, and later I secured samples which, although brittle, "rotten," or without cohesion of fiber, are in all respects similar to what I obtained by exposing a sheet of new, thoroughly first-class asbestos cloth to a moderate flame temperature for the space of five minutes.

The asbestos curtain at the Iroquois theater was an utter failure in three different ways :

1st, as already stated, it could not be lowered, and stuck fast after descending a distance variously estimated at from

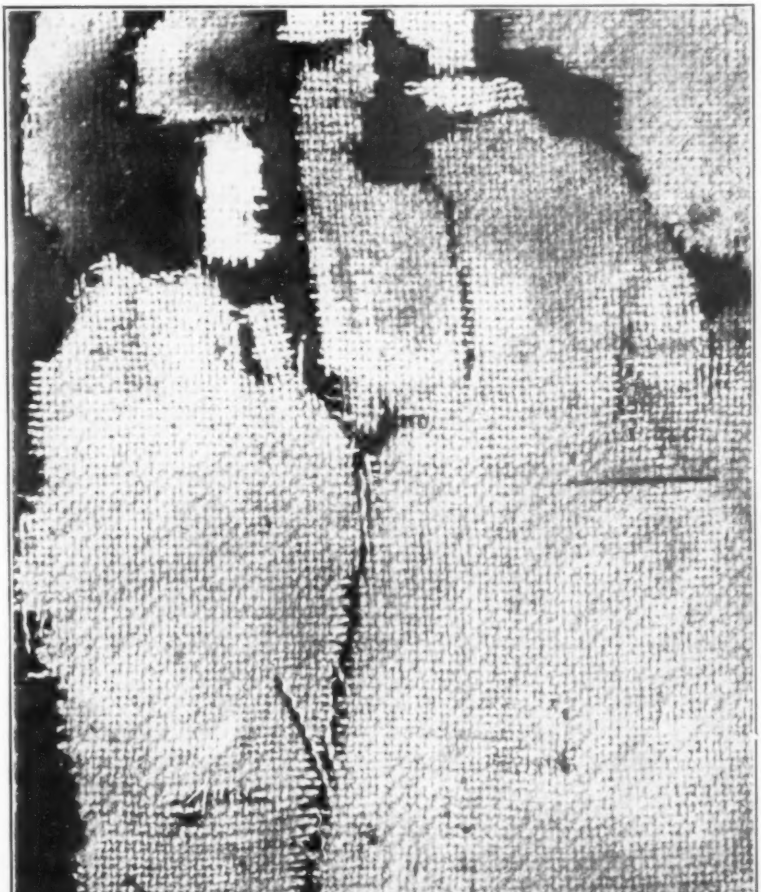

FIG. 8.—A SAMPLE OF THE IROQUOIS CURTAIN TAKEN FROM THE STAGE AFTER THE FIRE.
(From a photo.)

one-fourth to one-half the height of the proscenium arch.

2d, the Iroquois curtain was improperly hung, being supported at the top in part by being clamped between thin strips of pine wood about four inches in width by three-fourths of an inch in thickness. (So tolerant is the

public and so easy are public building inspectors, that I have myself seen in actual use in several theaters examples of an asbestos curtain hung from a batten of white pine to which it was nailed across the top.)

3d, the asbestos canvas of the Iroquois curtain, when exposed to actual fire, lost its strength and fibrous quality almost completely, and became so brittle that it would crumble under a very slight pressure, and became utterly incapable of withstanding the pressure of a strong draft of air, and too weak to hang up under its own weight.

CONCERNING ASBESTOS.

The word "asbestos" has become, in the public mind, a synonym for perfection in fire-proof material, but the investigations now to be described have made me believe that a simple asbestos curtain of even the very best quality will not form a durable and certain fire screen for the proscenium arch when exposed to a bad fire.

Any asbestos curtain may be expected to resist the ridiculously inadequate test of the flame of a gasoline torch, and any well hung asbestos curtain, *if it can be pulled down*, will probably endure longer than the brief period of two or three or four minutes, within which it should be possible to empty any theater; and meanwhile it might serve a most useful purpose in screening the flames from direct view.

In opposition to the failure of the Iroquois asbestos curtain we have an interesting test of action of asbestos curtain and smoke vents combined in the fire that destroyed the Girard Avenue Theatre in Philadelphia, on October 28, 1904, and which broke three hours after midnight on the stage when no one was present. On the arrival of the public fire department, three minutes after the first alarm, the flames were coming out of the skylight ventilators over the stage, which it is said were of one-eighth the stage area, and had opened automatically. The firemen at first found no fire or smoke in the auditorium, and the curtain hung there, and probably with the aid of the cool indraft toward the stage, kept flames out of auditorium for a period said to be fifteen minutes. Shortly after this the fire somehow passed into the auditorium; doubtless around the edge of the curtain or by the curtain becoming ruptured by falling material. It is curious to

observe how this case has been quoted as a triumph for the asbestos curtain, while the more important part played by the smoke vents was completely lost sight of!

While I regard this record as more of a triumph for the smoke vent than for the curtain, it is of great interest to note that under existing conditions, whatever they were as to quantity of burning scenery, this curtain, with the open smoke vents of one-eighth the area of the stage, lasted much more than long enough to have covered the escape of an audience. In all probability, this fire was much less fierce and rapid than the Iroquois and had far less scenery on the stage.

In the United States asbestos canvas costs anywhere from \$1.25 to \$3.50 per square yard, according to weight and texture, and a proscenium curtain of asbestos may cost anywhere from say \$175 to \$600.

In order to learn what difference there might be between different makes and grades of asbestos canvas, I obtained through various channels samples, each one or two yards square, from all of the prominent American manufacturers of theater curtains and also from each of the American manufacturers of asbestos cloth. I also cabled to London and had an architect familiar with theatrical work collect samples of asbestos curtain cloth none less than a yard square from the leading English manufacturers and dealers, under instructions to use every effort to procure some canvas that was woven from French or Italian or other than Canadian fiber.

When pressed hard for the pedigree of their samples, no one of these makers would furnish asbestos canvas under a guarantee that it was made from anything other than the Canadian fiber, and on chemical analysis, all of our specimens of canvas, obtained either at home or abroad, were found to be of a chemical constitution similar to that of the Canadian fiber.

The Canadian mineral is not the kind to which the name asbestos was first applied and, strictly speaking, is not true asbestos.

The Canadian asbestos is a fibrous crystalline variety of serpentine and contains about thirteen per cent. of water in chemical combination, plus a little hygroscopic water; whereas the form to which the name asbestos was first applied by the ancients contained no combined water whatever.

There are two or three minerals of very different chemical constitution which go under the name of asbestos:

1. Chrysotile, which contains about fifteen per cent. of water, twelve and nine-tenths per cent. chemically combined, and about two per cent. hygroscopic. This is essentially a silicate of magnesia.

2. Tremolite, which is anhydrous and is a silicate of lime and magnesia, with sometimes a little iron.

3. There is a mineral which is asbestiform in character, a silicate of iron and magnesia, known as anthophyllite.

The first named loses its strength at about six hundred and sixty degrees Centigrade, or just below redness, on the driving off of its water, but the last two, containing no combined water, stand more heat and are said not to fuse until about thirteen hundred degrees Centigrade, equal to twenty-four hundred degrees Fahrenheit, is reached. We did not measure this high fusing point. The behavior of some filaments in a blast lamp indicates a lower fusing point for the tremolite asbestos than as just stated on text-book authority.

The Canadian fiber is Chrysotile. This is now the common asbestos of commerce, and possessing in greater degree than the others the properties required for spinning and weaving, has come to be the only kind used in the manufacture of asbestos canvas.

The Georgia asbestos, although free from water in its chemical combination, and therefore not decomposing at low red heat, has for the most part a fiber too brittle for spinning, and is used for purposes not requiring strength of fiber.

The anhydrous Tremolite and amphibole asbestos are also found in Siberia and in South Africa, but all the anhydrous asbestos mined or quarried makes up an insignificant part of the asbestos of commerce, and although some of the cabinet specimens of anhydrous asbestos have long, silky, pliable fiber, I was unable to anywhere obtain cloth made from anhydrous asbestos.

Several kinds of asbestos canvas can be procured in the market. There is a distinction sometimes made in the trade between "absolutely pure" asbestos canvas, which contains no cotton, being made for filter cloth or other industrial purposes, and "commercially pure" asbestos canvas, which may contain from five per cent. to fifteen per cent. of cotton carded in with the asbestos fiber. These can be distinguished by picking a piece of the yarn into fine feathery condition and touching a match to the ends of the fiber and noting the flash and smell of burned cotton.

Certain manufacturers claim that a small percentage of cotton, besides facilitating the spinning and weaving into a strong, pliable canvas, improves the cloth for the purpose of painting a picture upon it, as for a drop curtain, and claim that this small amount of cotton does not impair its fire resistance. Asbestos fibers are very slippery and difficult to card and spin, and by taking advantage of the spiral structure of the relatively few cotton fibers to bind the asbestos fibers together, the process of manufacturing a smooth canvas is greatly facilitated.

A third variety of asbestos cloth which has been highly recommended (on *a priori* grounds rather than from tests), by fire chiefs and architects for a theater curtain, contains very fine brass wires, of No. 33 and No. 34, standard gauge, or only about the $\frac{1}{150}$ part of an inch in diameter, woven in with the asbestos yarn. My tests proved that *these fine wires add nothing to the strength of the heated canvas.* The wire used was found by analysis two-thirds copper, one-third zinc, with a trace of lead, perhaps two per cent.; this analysis proving it to be an ordinary brass wire. Probably the extreme fineness of the wire used and the quick oxidation or volatilization of the zinc is a cause of its weakness when heated.

All of the alleged asbestos curtains that I have seen have really been of the ordinary commercial asbestos, and I regard the stories of painted burlap masquerading as asbestos in theater curtains as mostly idle talk.

Test of Asbestos Canvas.

Since my experiments on the effect of heat upon the tensile strength of asbestos cloth and asbestos fiber quickly disclosed that the ordinary commercial asbestos lost its strength at a heat just below redness, sufficient to drive off the combined water, in order to be sure of our ground, I had three independent series of tests upon asbestos made by three different experts, and by very different methods, myself laying out only the outlines of the test desired and leaving the observations and reports to the respective experts. The results of all three tests proved independently that the character of asbestos cloth as to resisting a high degree of heat is utterly different from what is popularly supposed.

1*st Series.*

The first series of tests were made by Prof. Charles E. Fuller, in the Mechanical Engineering Laboratory of the Massa-

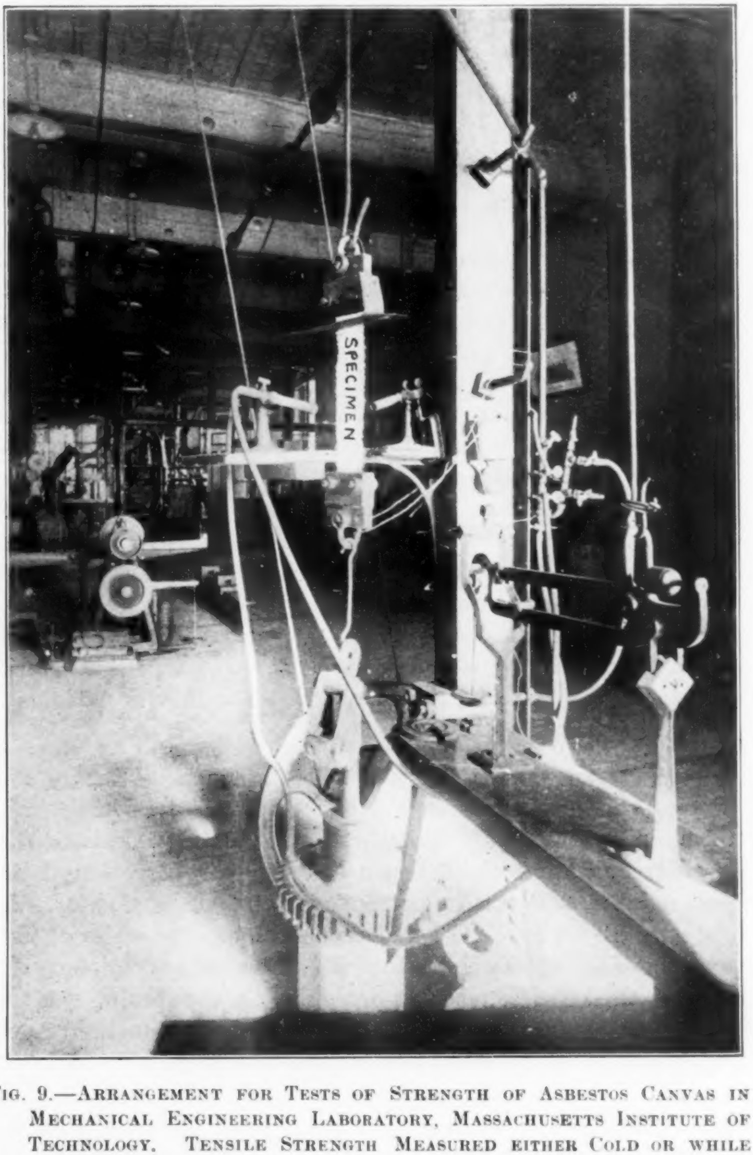

FIG. 9.—ARRANGEMENT FOR TESTS OF STRENGTH OF ASBESTOS CANVAS IN MECHANICAL ENGINEERING LABORATORY, MASSACHUSETTS INSTITUTE OF TECHNOLOGY. TENSILE STRENGTH MEASURED EITHER COLD OR WHILE HEATED BY COMMON GAS FLAME OR BY BLAST LAMP FLAME TO VARIOUS DEGREES.

chusetts Institute of Technology, on a special testing machine designed for measuring the strength of sailcloth, and which had been previously used in a series of tests for the United States Government. We had this newly fitted with double Bunsen gas burners, as shown in Fig. 8, so arranged that a specimen could be either tested cold or tested for strength while heated to any desirable degree in either an ordinary gas flame or heated up to moderate redness in the flame of the blast lamps. These tests were made with great care, repeating the test on two or more specimens in almost every case and with a degree of attention to detail which I have not space here to set forth. I believe the results absolutely reliable.

In brief, *we found that every one of these specimens of asbestos canvas, English, French and American alike, when heated for from two to five minutes to a little below redness in a common gas flame, or barely to redness in the Bunsen flame, lost from sixty per cent. to ninety per cent. of its strength*, and that the fiber became very brittle.

We were surprised to find that the samples with the wire insertion, when tested hot, were no stronger than the samples without wire. On cooling, they regained a little of the strength due to the wire.

I have condensed the results of these tests into the tables which follow on pages 110–113.

At time of weighing and measuring thickness of American samples, relative humidity of air was 80 per cent., temperature, 68 degrees F. When weighing and measuring foreign samples, relative humidity, 25 per cent., temperature, 71 degrees F. Thickness measured between glass plates, 2 inches square, under 8 lbs. pressure. Specimens prepared by cutting strips $2\frac{1}{2}$ inches wide and ravelling out threads carefully from edge until width of 2 inches to the nearest thread was left.

All specimens held with 12 inches between the grips.

Load applied, in nearly all tests, by stretching the cloth at rate of $\frac{1}{2}$ inch per minute. A few tests were run at speed of stretching at rate of $\frac{1}{5}$ inch per minute for comparison, and no difference found in strength between these rates of speed.

The results of test of strength while hot, given in the table, are averages from three specimens. These three different trials gave substantially uniform results.

TABLE 1.—EFFECT OF HEAT UPON STRENGTH OF ASBESTOS CANVAS (FOREIGN SAMPLES).

	Plain cloth without wire.				With fine wire spun in yarn and woven in.			
Letter marked on sample	B	C	D	F	A	E	G	H
Name of manufacturer	Germain Boisne, Freres et Cie, France.	Germain Boisne, Freres et Cie, France.	Germain Boisne, Freres et Cie, France.	Leyland & Birmingham Rubber Co., England.	United Asbestos Co., Ltd., England.	Merryweather & Co., England.	Leyland & Birmingham Rubber Co., England.	Bell's Asbestos Co., Ltd., England.
Equiv. selling price (in London) per sq. yd. not made up into curtain*	$1.50	$0.45	$0.55	$0.37	$1.25	....	$0.60	$0.90
Actual weight of cloth per sq. yd. in lbs	3.00	1.80	1.71	1.98	2.52	2.88	3.00	2.43
Average thickness in inches (between glass @ 2 lbs. per sq. in.)	.072	.044	.068	.058	.069	.075	.079	.059
Statement of per cent. of cotton contained	0.	3.	3.	0.	....	0.	....	....
Per cent. loss of weight found after heating bright red (mean of 2 samples)	13.6	17.6	28.2	16.5	20.1	20.9	11.9	17.4
Probable actual per cent. of cotton	0.	1.6	14.	2.5	6.	7.	0.	3.4
Warp: No. of warp threads per inch	27.	41.	27.	30.	36.	35.	27.	41.
Warp: " " strands in asbestos yarn	2.	2.	2.	2.	1.	1.	1.	2.
Warp: " " brass wires in each "	0.	0.	0.	0.	1.	1.	1.	2.
Warp: Diameter of wire in inches	....	....	....	....	.0076	.0062	.0062	.0061
Warp: Breaking load 1 wire, cold in lbs. (mean of 3)	....	....	....	....	3.12	1.97	2.22	1.88
Warp: Total strength of wire per inch in width in lbs	....	....	....	....	56.	34.5	29.8	37.6
Filling: No. of filling threads per inch	32.	35.	22.	27.	19.	19.	22.	26.
Filling: " " strands asbestos yarn	2.	1.	2.	2.	1.	1.	1.	1.
Filling: " " " brass wire	0.	0.	0.	0.	1.	1.	1.	1.
Filling: Diameter of brass wire in inches	....	....	....	....	.0064	.0061	.0062	.0061
Filling: Breaking load 1 wire cold in lbs	....	....	....	....	2.30	1.90	2.10	1.90
Filling: Total strength of wire per inch in width in lbs	....	....	....	....	22.	18.	23.	24.5
Stretch of Cloth at Maximum Load:								
Per cent. of elongation lengthwise (mean of 2)	15.6	15.3	13.9	13.0	14.	14.5	16.1	12.5
" " " " crosswise	8.1	10.3	6.8	6.0	4.6	6.2	10.9	6.3
Strength of Cloth Tested Cold (in lbs. per inch in width).								
Crosswise—Filling	119.	56.	50.	20.	49.	42.	53.	53.
Lengthwise—Warp (mean of 2)	130.	112.5	49.	81.	72.5	97.5	113.5	128.
Strength of Cloth While Heated (in lbs. per inch in width). All of following samples tested lengthwise (warp).								
Heated 1 side with a blast lamp 1 min. Dull red—broken while hot	31.5	16.5	7.7	22.3	3.	11.2	18.8	16.
" " " " " " 5 " " " " " "	17.5	14.1	8.8	13.9	4.0	9.0	12.4	16.3
Heated both sides with blast lamps 5 min. Bright red—broken while hot	18.4	14.8	7.0	14.8	2.3	9.9	11.5	9.7
Heated 1 side in common gas flame 5 min. Broken while hot	60.2	34.5	9.0	36.8	11.2	21.2	36.5	32.
" " " " " " 15 " " " "	44.0	18.5	8.3	25.0	8.5	14.0	30.2	24.

TABLE 1.—*Continued.*

Strength after Cooling Off (in lbs. per inch in width).								
That which had been heated dull red 5 min	16.8	12.4	7.7	12.4	31.5	9.8	15.2	42.5
" " " " " bright red 5 min	16.8	11.9	5.9	12.4	10.8	10.4	12.8	13.5
Percentage of Original Strength Lost—Tested while Heated.								
Heated dull red 1 min	76.	85.	84.	72.5	96.	88.5	83.5	87.5
" " " 5 "	86.5	87.5	82.	83.	94.5	91.	89.	87.
" bright red 5 min. (2 sides)	86.	86.	86.	82.	97.	90.	90.	92.5
" in common gas flame 5 min	54.	69.5	81.5	54.5	84.	78.	68.	75.
" " " " " 15 "	66.	83.5	83.	69.	88.	85.5	73.	81.
Per Cent. of Original Strength Lost—Tested after Heating and Cooling Off.								
Heated dull red 5 min	87.	89.	84.5	85.	57.	90.	86.5	67.
" bright red 5 min	87.	89.5	88.	85.	85.	89.5	89.	89.5

* United States Custom House duty, 25 per cent.

Samples B and F were practically free from cotton, C contained a small amount, and A, E, D and H contained considerable cotton.

Samples A, B, C, D and H are probably the only specimens that can be relied on as coming from the original manufacturers. The firms supplying E, F and G are dealers only.

A, F, G and H are understood to be of Canadian asbestos; E is probably of Canadian fiber; B, C and D are probably a mixture of Canadian and European fiber.

Sample B is called "chemically pure cloth"; C is called "commercially pure cloth."

Sample E is, as supplied, to pass inspection of the London County Council.

TABLE 2.—EFFECT OF HEAT UPON STRENGTH OF ASBESTOS CANVAS (AMERICAN SAMPLES).

Specimens arranged in order of weight or supposed strength of fabric.

CONSTRUCTION.	Plain cloth without wire.			With fine wire spun in yarn and woven in.				
Number marked on sample	A	B	C	D	E	F	G	H
Name of manufacturer	Trainer	Johns	K & M	K & M.	Trainer	Johns	K & M	K & M
Trade name or number	"Standard."	No. 902 Medium.	No. 9 Plain	No. 13 A "Metallic."	"Wire Insertion."	No. 6193	No. 13 B "Metallic."	No. 13 C "Metallic."
Approx. ordinary selling price per sq. yd. of cloth (not made up into curtain)	$1.25	$2.75	....	....	....	....	....	....
Actual weight of cloth per sq. yd. in lbs.	2.07	2.25	3.06	1.89	2.16	2.70	2.88	3.51
Average thickness in inches (Between glass at 2 lbs. per sq. in)	.073	.069	.077	.045	.046	.059	.064	.078
Manufacturer's statement of per cent. of cotton contained	5%	....	5%	10	....	....	10	....
Per cent. loss of weight found after heating bright red mean of two specimens	22.6	14.	19	19.8	14.4	18.5	20.1	14.7
Probable per cent. of cotton	8.6	0	5	5.8	0.4	4.5	6.1	0.7
Warp: No. of warp threads per inch	12.5	12.5	20	24	24	14.5	20	20
Warp: " " strands in asbestos yarn	2	2	2	1	1	2	2	2
Warp: " " brass wires in each yarn	....	....	....	1	1	2	2	2
Warp: Diameter of wire in inches	....	....	....	.0060	.0062	.0062	.0055	.0080
Warp: Breaking load 1 wire cold in lbs. (mean of 2)	....	....	....	2.10	2.02	2.02	1.60	3.35
Warp: Total strength of wire per inch in width	....	....	....	50	44	29	32	67
Filling: No. of thread per inch	9.5	9.7	8.5	14	13	9.5	9.5	8.5
Filling: " strands asbestos yarn	2	2	2	1	1	1	2	2
Filling: " " brass wire	....	....	....	1	1	1	1	1
Filling: Diameter of brass wire in inches	....	....	....	.0055	.0062	.0062	.0055	.0080
Filling: Breaking load 1 wire cold in lbs	....	....	....	1.60	1.90	2.45	1.60	3.20
Filling: Total strength of wire per inch in width in lbs	....	....	....	23	25	22	15	27
Stretch of Cloth at Maximum Load:								
Per cent. of elongation lengthwise (mean of 2)	13.7	11.6	15.6	13.1	10.1	10.8	14.2	18.8
" " " crosswise (mean of 2)	5.3	4.1	4	6	6	6.1	6.6	6.7
Strength of Cloth Tested Cold (in lbs. per inch in width):								
Crosswise filling	30	32	45	58	46	45	76	84
Lengthwise warp	43	68	108	80	97	133	173	212
Strength of Cloth While Heated (in lbs. per inch in width). All of following samples tested lengthwise (warp):								
Heated 1 side in common gas flame 5 min. (did not appear red hot)	9	20	30	14	19	17	25	42
Heated 1 side in common gas flame 10 min. (did not appear red hot)	10	23	27	13	14	25	27	37
Heated both sides in common gas flame 5 min. (did not appear red hot)	8	30	17	13	16	17	20	39
Heated to dull redness by blast lamp 1 to 2 min. broken while hot	6	19	17	11	14	15	18	18
" " " " " " " 5 min. " " "	8	19.5	16	9	14	15	17.5	20
" " bright redness by blast lamp 5 min. " " "	8	14.5	14.5	7	9.5	12	12.5	12
" " " " " " " 15 " " " "	7	14	14	8	11	14	11	12

TABLE 2.—*Continued.*

Strength After Cooling Off (in lbs. per inch in width):								
That which had been heated dull red 5 min.	7	14	10	14	25	41	35	73
" " " " " bright red 5 min.	8	13	12	13	14	31	25	51
Percentage of Original Strength Lost. Tested While Heated:								
Heated common gas flame 1 side 5 min.								
" " " " " " 10 "	79	60	77	83	83	85	85	81
" " " " 2 sides 5 min.								
Heated to dull redness 1 min.	86	72	84	86	86	89	90	92
" " " " 5 "	82	71	85	89	86	89	90	91
" " bright redness 5 "	82	78	86	91	90	91	92	95
" " " " 15 "	84	80	87	90	89	89	94	95
Percentage of Original Strength Lost. Tested After Heating and Cooling Off:								
Heated dull red 5 min.	84	80	91	83	75	69	80	68
" bright red 5 min.	82	81	89	84	86	77	85	77

Specimen B was the only one of the above samples that when heated to redness in a glass ignition tube in Professor Warren's tests showed no organic material. Also in Professor Fuller's tests this showed no burning out of cotton or other organic matter when the flame of blast lamp was applied.

G and E also showed but little sign of organic material when the blast lamp was applied. The difference between sample B, which showed in one experiment 13.5% and in another 14.5% loss on ignition, averaging 14.0% of water, and the others may be taken as showing the amount of cotton present, for all of the above samples were found by the tests of the mineralogist to be made from the chrysotile asbestos.

The temperature and relative humidity were frequently observed during the tests. The temperature was about 70 to 72° F., and the relative humidity averaged 77 to 97%.

The following tests were made on strength of samples of asbestos twine, cord, and rope, manufactured by the H. W. Johns-Manville Co.:

TABLE 3.

Material.	Description.	No. of Specimen.	Breaking Load.		Elongation at Maximum Load. Measured Length.	Elongation.
Three-eighth-inch rope.	3 strands, each made up of 8 threads of plain asbestos with a manila core. Each thread made up of 2 strands of yarn, of which 15.5 ft. in length weighed 1 pound.	11-1	588 lbs.	broken cold.	3 inches.	18.7%
		11-2	600 lbs.	broken cold.		
		11-3	108 lbs.	broken hot after heating dull red 5 minutes.		
		11-4	81 lbs.	broken hot after heating bright red 5 minutes.		
One-quarter-inch rope.	3 strands, each strand made up of 4 threads of plain asbestos. Each thread made up of 2 strands of yarn, 67.5 feet per pound.	12-1	330 lbs.	broken cold.	3 inches.	14.6%
		12-2	45 lbs.	broken hot after heating to dull red 5 minutes.		
		12-3	37 lbs.	broken hot after heating to bright red for 5 minutes.		
One-eighth-inch cord, No. 804.	4 threads of plain asbestos, each thread consisting of 2 strands of yarn, of which 130.5 feet weighed 1 pound.	13-1	143 lbs.	broken cold.	12 inches.	7.3%
		13-2	132 lbs.	broken cold.		
		13-3	124 lbs.	broken cold.		
		13-4	14 lbs.	broken hot after heating to dull red 5 minutes.		
		13-5	18 lbs.	broken hot after heating to bright red for 5 minutes.		
		13-6	35 lbs.	broken hot after heating in common gas flame 5 minutes.		
Plain asbestos sewing twine.	2 strands plain asbestos yarn, of which 1,364 feet weighed 1 pound.	14-1	12.8 lbs.	broken cold.		
		14-2	11.3 lbs.	broken cold.		
		14-3	12.0 lbs.	broken cold.		
		14-4	1.1 lbs.	broken hot after heating bright red 2 minutes.		
		14-5	1.0 lb.	broken hot after heating bright red 5 minutes.		
		14-6	0.8 lb.	broken hot after heating in common gas flame 5 minutes.		
Sewing twine with wire insertion.	2 strands of plain asbestos yarn, 2 strands of brass wire; diam. .0070 inch; 833.3 feet per pound. Breaking strength of 1 wire cold, 2.12 pounds.	15-1	17.0 lbs.	broken cold.		
		15-2	19.5 lbs.	broken cold.		
		15-3	16.5 lbs.	broken cold.		
		15-4	2.1 lbs.	broken hot after heating bright red 2 minutes.		
		15-5	1.9 lbs.	broken hot after heating bright red 5 minutes.		
		15-6	2.3 lbs.	broken hot after heating in common gas flame 5 minutes.		

It will be seen from the above that the asbestos cord was affected by heat very much as the canvas was, and lost nearly all of its strength after brief exposure to dull red heat, or even to a heat a little below.

2d Series.

I desired tests on larger sheets of the canvas, more nearly reproducing the conditions of use, and so the tests of our second series were made at the Underwriters' laboratory, in Chicago, by constructing asbestos curtains about six feet square and testing them with the same furnace and apparatus that had been provided in the yard attached to this laboratory for testing fire shutters and fire doors. Unfortunately, we found the furnace in poor working condition because of a temporary defect in the gas supply, such that we could not regulate the temperature evenly over the entire curtain or measure it precisely by the pyrometer. After testing several curtains we therefore suspended these tests.

The large sheets under these conditions made a much better showing than the small samples had made in the laboratory. Nevertheless these Chicago tests fully confirmed the conclusion, derived from our Boston tests, that asbestos cloth is rapidly weakened by the heat of an ordinary fire to an extent that makes a curtain composed wholly of asbestos cloth an unreliable fire screen for the proscenium arch of a theater, if expected to endure more than a few minutes; and it was proved that the asbestos canvas was so weakened that it would be ruptured easily by a blow from any falling material or by a strong current of air. It was noted during these tests that the Johns curtains were found particularly weak in their horizontal threads, permitting vertical rents to be easily made. That the canvas of the other makers is similarly much less strong horizontally than it is vertically, or in the warp threads, may be seen from the Boston tests set forth in the previous table. The seams sewed with asbestos thread showed no special weakness more than the canvas.

A notable feature in these furnace tests with those curtains that contained from five per cent. to eight per cent. of cotton was the flame that played all over the outer face of the cloth when the furnace was lighted, and which might be disquieting to an audience in giving for a moment an impression that the asbestos curtain was burning up.

3d Series.

For the third series of tests, the friendly services of Prof. William Otis Crosby, of the Massachusetts Institute of Technology, in charge of the Department of Economic Geology, and Dr. C. H. Warren, Professor of Mineralogy in the same institution, were

enlisted to examine all of the varieties of asbestiform minerals found in the extensive cabinets of the Institute of Technology and the Boston Society of Natural History, in the hope of finding specimens from some locality that possessed all the qualities properly attributed to asbestos. The result of this search, in brief, was that nothing was found possessing characteristics materially different from the hydrous Canadian fiber on the one hand, and the anhydrous fiber from Georgia on the other. Specimens of asbestos of the first class lost their strength at a heat which drove the water off; the specimens of the second class had fibers that were too stiff or too brittle for spinning and weaving, or they were reported as occurring in quantity too small for commercial purposes.

Tests were next made by Professor Warren to learn the precise degree of heat required to injure the strength of asbestos fiber.

Upon testing specimens of the Canadian fiber by heating it in a platinum crucible within a clay cup and within a coil heated electrically, raising the temperature slowly, weighing the specimen repeatedly, and all the time measuring the temperature in the crucible by electrical methods, it was found that a temperature up to 250° Centigrade, equivalent to 482° Fahrenheit, caused no driving off of the water chemically combined, and no apparent change in the pliability or strength of the fiber.

A heat just below dull redness proved to be the critical point. One-half hour at from 440° C. to 480° C., equivalent to 850° F., drove off about three per cent. of the combined water and made the fiber slightly more brittle than at first, with some loss of natural luster.

Heated to from 630° to 650° C. for five minutes, averaging 1,152° F., eleven per cent. of the water is driven off, and the fiber becomes slightly brown and very brittle and crumbly.

Heated for only thirty seconds to 750° C., the fibers of Canadian asbestos lost their cohesion.

A faint red heat corresponds to about 1,100° F. (The Austrian engineers, from their experiments on a theater model in 1885 already referred to, concluded that a temperature of 800° C., equal to 1,472° F., would be reached in a fire on a stage crowded with scenery.)

In other trials Professor Watson found that a piece of asbestos canvas a foot square lost its strength badly, so that it could be torn between the fingers, after it had been held five minutes

against a moderate wood fire that did not heat it to visible redness, and therefore probably not to 650° C.

From a series of such tests Professor Warren concluded that a theater curtain made of the Canadian or chrysotile asbestos fiber alone could not be expected to hold together for more than a few moments if a temperature of 650° C., equivalent to about 1,200° F., was reached, and calls attention to the fact that, being a non-conductor, the asbestos canvas would arrest and absorb the radiant heat from the burning scenery and have its own temperature rapidly raised.

In the course of sundry other tests he found independently that the fine brass wire inserted in certain samples of the canvas added practically nothing to their strength while hot. Wires pulled out from the canvas and held in an open Bunsen flame lost their strength instantly. This led Professor Warren to suggest that iron wire of, say, twenty-five or twenty-eight gauge, would prove a much greater addition to the strength of curtain cloth, but in the Chicago furnace tests I noted that iron wire, about No. 18, was rapidly oxidized in the gas flame.

The asbestos residue left after driving off the water is practically infusible and would doubtless adhere together, and being supported by the iron wire would form an effective screen long enough to permit escape and to restrain the progress of the flames while the firemen were coming. Wrought iron is less readily fusible than steel, but the steel wire would probably hold up to, say, 900° C., or 1,650° F. Steel melts at 1,200° to 1,300° C.; pure iron about 1,600° C.

I found the small iron stairway over the Iroquois stage showed effects such as are produced by red heat. Glass in the skylights over the Iroquois stage was fused, which indicates about 875° to 900° C., or 1,650° F.

Steel Plate Protected by Asbestos Material.

In our Chicago furnace tests we also experimented upon sundry combinations of asbestos, asbestos felt and asbestic cement, with thin steel plate and combined with wire netting, the asbestos being placed on the stage side in the hope that it might shield the steel from the full heat and thus prevent it showing red hot on the auditorium side, while the steel would give strength. We had to suspend these tests because of some temporary trouble with

the gas supply to the furnace, but they were carried far enough to prove an endurance more than ample for their purpose as a shield while the audience is escaping, and it was plain to all who witnessed these tests that the sheet steel curtain, protected with some asbestic material on the fire side, possessed far greater strength and endurance against fire than the simple asbestos. The thin sheet of steel, moreover, cut off the view of the fire that was apparent through the texture of the asbestos canvas.

With care given to the design of the guides and fastenings at edges and top, so that after it was lowered the curtain could not be pulled out by warping, buckling, "smoke explosions" or pressure of air, the steel curtains would have a value to the fire underwriter that no asbestos curtain can possess, and would probably hold a fire on the stage from entering the auditorium.

The general type of steel proscenium curtain finally adopted in Chicago and required at all theaters was worked out somewhat hurriedly, according to the average judgment of the Aldermanic Committee, in advance of any other tests than the failure of the Iroquois curtain.

It consists of a light framework of steel angle irons with corrugated plate about one-sixteenth inch thick on the auditorium side, and some asbestic non-conducting material on the stage side, with an air space of one, two or three inches between. Where guided only by loops on stationary vertical cables at its vertical edges, it is required to lap over the edge of the arch about eight inches. A curtain structure of this kind of the ordinary size weighs from two tons to six tons.

The hanging of most of these Chicago steel curtains would be improved by more substantial iron channels to hold the edge, and by the addition of positive down-haul tackle or some arrangement by which the counterweight could be thrown off, for now the great weight of these curtains is so nearly counterpoised that it is entirely possible that the excess of air pressure against its surface of about one thousand square feet may prevent the slight excess of gravity from lowering it. The Austrian experiments and the story of some of the eye-witnesses of the Iroquois disaster indicate the possibility of a strong outward air pressure from the expanding air.

Finally, regarding fire curtains, it should be said that with proper smoke vents and complete automatic sprinklers, the perfection of the curtain becomes of lesser importance.

THE FIREPROOFING OF SCENERY.

Theatrical scenery is ordinarily painted on a strong linen canvas weighing about six and six-tenths ounces per square yard. Heavy cotton sheeting is sometimes used for the cheaper temporary productions. The gauze used for skies and transformation scenes is of cotton, of texture like mosquito-netting. Frames and battens and profile backings are of white pine. The canvas is first stretched on a frame and stiffened by a coat of glue size applied warm with a broad brush. Next, it receives a priming coat of whiting and glue size, and is then ready for the scene painter. The mineral colors used are mixed with water and glue, and many tests prove that the painted canvas is somewhat less readily combustible than the unpainted, and that the heavier is the coat of pigment the more flame is retarded.

The fireproofing of scenery canvas and other cloths and fabrics has from time to time, during the past fifty years, engaged the attention of many talented men, and one who now consults only the articles in books and technical pamphlets is led to believe that this fireproofing of canvas or cloth can be accomplished by brushing the surface over with either one of several solutions of chemicals.

After reviewing whatever I could find in print, after consulting with several experienced scenic artists, and after making tests myself, and later enlisting the friendly assistance of several experienced chemists to carry on independent investigations of all solutions prominently recommended for the fireproofing or flame-proofing of fabrics, I regret to conclude:

> 1st, that the best that is possible in the fireproofing of scenery is far from satisfactory;
>
> 2d, that the petty tests that have satisfied certain distinguished chemists are very misleading as guides to what will happen when the same process is tested on the larger practical scale;
>
> 3d, that the best we can hope to accomplish is to "flame-proof" a fabric so that it will not ignite from a match, an electric spark or a gas jet; or so that if ignited it will not burst into flame.

This much of protection, little and disappointing as it is, is of great value and worth all that a good process costs, if it can be

accomplished in practice without injury to fabric or colors; for if we can thus prevent the little flame from quickly spreading, we have removed perhaps nine-tenths of the danger of a fire starting on the stage, but it falls far short of what many have believed and loudly proclaimed was within easy reach.

Inefficiency of All Methods of Fireproofing Scenery.

Once get the gauze and canvas and pine on the stage enveloped in flame, everything "fireproofed" will burn to total destruction with substantially as great a rush of flame and suffocating smoke as with the untreated material. Indeed, the chemicals may make the fumes worse.

After having investigated the question of fireproofing the scenery faithfully, probably with greater thoroughness than has ever been done heretofore, I am led to believe that we must after all rely on the safeguards of the engineer rather than those of the chemist, for the safeguarding of human life in theaters.

The efficient fireproofing of the great quantity of white pine used in frames, battens and profiles (eight thousand square feet in the case of the Iroquois) appears to be a practical impossibility. The eleven miles of manila ropes cannot be "flame-proofed" without too great a sacrifice of their strength.

New Tests and the Theory of Action of "Fireproofing" Chemicals.

Distrusting the ordinary test of trying to ignite a small strip of the treated cloth with a match or gas flame, early in these studies I took about one-fourth a square yard of cloth treated with phosphate of ammonia, which is the most efficient fire retardant of all the half hundred chemicals and mixtures yet recommended, and hanging it in sheets half an inch apart within a piece of common stove pipe two feet long, lined with a sheet of asbestos in order to check the loss of heat by absorption in the cold metal, lighting it at the bottom with a little wad of "excelsior" wood shavings, I saw this "most perfectly fireproofed cloth" disappear with a rush of flame in forty seconds.

The difference in the results of a test of a single strip over a gas flame and a series of parallel sheets hung near together within a chamber is that in the second case we confine the radiant heat and the distilled combustible gases very much as they would be

confined in the closely hung sheets of scenery over the stage (see Figs. 1 and 2).

It is almost inconceivable that any of the various solutions used for fireproofing canvas or wood could so change the cellulose, gums and resins, of which these mainly consist, as to prevent their destructive distillation with the evolution of the same volume of inflammable gas, much like ordinary illuminating gas, that they ordinarily give out when heated to the char point.

We may find it slow work to light the wet twigs and green wood for our camp fire, but once well started, they burn furiously. The best fireproofing of fabrics amounts to about the same as substituting a fabric of wool for one of flax or cotton. Woolen mills by fifty years' experience are no better insurance risks than cotton mills.

Our various tests, interpreted from the scientific point of view, indicate that about all that we can hope for from the application of "fireproofing" chemicals is as follows:

> 1st, the destructive distillation of the chemical may keep the surface of the cloth near the applied flame bathed for a few seconds in a thin film of steam or inert gas, arising from the distillation of the microscopic quantity of the chemical lodged in the fabric, thus keeping the oxygen of the air away from the carbon for a moment.
>
> 2d, the dissociation or distillation of this little quantity of the chemical absorbs a little of the heat applied or evolved.
>
> 3d, the chemical used may have a non-volatile base which will fuse and cover the surface of the combustible carbon with a glassy film that, although exceedingly thin, will keep this carbon beyond reach of the oxygen of the air. This may perhaps lock up from twenty-five per cent. to fifty per cent. of the heat-giving content of the fiber.

I have already intimated that phosphate of ammonia has given the best record in fire-retardant quality of any of the many chemicals and mixtures tested. Theoretically, we should expect it to do so, for its chemistry fulfils the above conditions. First, it has a little tendency to gather dampness, and to dry this out absorbs a little heat. Next, as the heat rises, ammonia is given off, and the thin film of this repels the oxygen of the air. When the ammo-

nia is gone we have left the otho phosphoric acid which in liquid form covers the surface and preserves it from oxidation under increasing heat. At 300 to 400 degrees Fahrenheit this decomposes, giving off water; at higher temperatures, it gives off its remaining water. In all of this dissociation it absorbs some heat until we have left, at full red heat, fused meta-phosphoric acid *as a liquid film surrounding the fixed carbon* remaining from the destructive distillation.

On the other hand, the phosphate of ammonia has its disadvantages. A manufacturing chemist, perhaps of widest experience of any in this country in the practical chemistry of the phosphates, warns me that for its best efficiency it must be applied in a strong or saturated solution, but if very strong, it may in time disastrously affect the strength of the fiber, that it is somewhat deliquescent, has a tendency to develop fungous growth, that in time it may part with a portion of its ammonia, becoming the acid ammonium phosphate which has a tendency in presence of moisture to attack metals, while in a warm atmosphere the free phosphoric acid attacks some colors.

The foregoing cautions by my friend, the chemist, were derived from experience on other material than stage scenery, and we shall soon have plenty practical experience to show if phosphate of ammonia is injurious to scenery, under the practical conditions of use, for this has been used during the past year and a half more than any other substance to meet the enforcement of the laws of certain cities requiring all stage scenery to be fireproofed. The diluteness of the solution that has been applied in some instances within my observation will tend to lessen its injurious qualities in the same degree that it weakens its flameproofing, and the tendency of any antipyrine to promote mildew in damp atmospheres can probably be prevented by adding some antiseptic or germicide to the solution.

History and Practice of Fireproofing Canvas.

After each of the great historic theater fires that have occurred since the science of chemistry was born, this subject of fireproofing cloth has been studied by chemists of eminence, and nearly all of the chemicals and compoundsr ecently brought forward by scenic artists and dealers in painters' supplies are the same that have been recommended over and over again for the past fifty years.

The tests made from time to time for proving their efficiency have not copied practical conditions.

It is said that fifty years ago, after a serious fire in the Berlin Opera House, it was made the custom to soak the scenery canvas in a strong solution of alum; nearly fifty years ago a Parisian chemist carefully examined the subject of fireproofing scenery, and orders are said to have been issued that all stage scenery be impregnated with silicate of soda. Fifty years ago the value of phosphate of ammonia was recognized as an antipyrine. Forty-five years ago an elaborate series of researches was reported to the British Association for the Advancement of Science, embracing a great range of chemicals, with many tests for determining the most effective strength of solution to be applied. Nearly thirty years ago, after the Brooklyn Theater horror, some of the scenery in Wallack's Theatre in New York is said to have been fireproofed with tungstate of soda, and the well-known New York chemist, Dr. R. Ogden Doremus, called the attention of American theater managers to phosphate of ammonia. More than twenty-five years ago a committee of the British House of Commons took testimony on this matter of fireproofing scenery, and the manager of the Criterion Theater testified that he regularly used sodium tungstate in the preparation of new scenery. Curiously, our recent tests fail to show any great virtue in sodium tungstate as an antipyrine. Twenty years ago the London Society of Arts reported on fireproofing of stage scenery and reported that the scenery in nearly all London theaters was treated with some fire retardant preparation.

Twenty years ago a committee of the Franklin Institute of Philadelphia studied and reported on this subject, recommending sundry chemicals.

Eleven years ago Prof. Thomas H. Norton devoted to this subject his presidential address before the Section of Chemistry in the American Association for the Advancement of Science at the Brooklyn meeting, and made it appear that fireproofing of fabrics was easy.

Nevertheless, it is probable there was at the time of the Iroquois fire hardly a piece of scenery on a theater stage in the United States or England, or anywhere else, that had been subjected to fireproofing treatment.

The veteran manager, John B. Shoeffel, tells me that from his experience with the French and English made scenery used in

the American tours of Bernhardt, Rejane, Mounet-Sully, Coquelin, Mary Anderson, Irving and others under his management, it is his confident belief that none of it was fireproofed. His experienced stage mechanic, William J. Kelly, confirms this and says further, that according to his personal experience on the stage of several London theaters, none of their scenery was fireproofed. The eminent scenic artist, Walter Burrage, of Chicago, tells me that through personal experience in England and conference with scene painters from the Continent, he has found there was no general use in Europe of fire retardant solutions in the preparations of scenery.

A year ago Mr. E. O. Sachs, Secretary of the British Fire Prevention Committee, wrote me that there was then no requirement for the fireproofing of scenery by chemical solutions in the English law, and in his compilation of the Building Laws of European cities, in Vienna alone, do we find mention of fireproofing of scenery, and there very vaguely.

Thus, notwithstanding widespread belief, backed by much eminent authority that scenery could be readily flame-proofed, it has not been done.

Why Stage Scenery Has Not Been Flame-proofed.

In brief, the reasons are:

1st, it adds to the cost by an amount that may be estimated at from $250 to $500 for the average five-act drama, having 25,000 square feet of canvas, and adds two or three times this cost for a great spectacular piece. Seldom would flameproofing add more than five per cent. or ten per cent. to the cost of an outfit of scenery.

2d, there is a fear that most of the fireproofing chemicals injure the strength of the canvas.

3d, the scenic artists have feared the effect on their delicate colors.

4th, some of the chemicals proposed tend to rust and loosen the iron fastenings and tacks.

5th, most of the stage scenery in existence is traveling around the country, stopping only a brief time in one city, and it is a tedious matter for the local authorities to make certain that it has been fireproofed.

6th, the appalling theater catastrophes have come almost a

generation apart. The people and the officials have short memories for their lessons. Inspectors become easy about special laws which, passed under pressure of a great calamity, soon become dead letters.

7th, the general public is thoughtless and indifferent and runs its chance.

Therefore, at the present time, although since the Iroquois fireproofing has become a general rule, remembering the likelihood that in future as in the past the fireproofing of scenery will become neglected, we may all the more emphasize the importance of the perfected automatic smoke vent and of the automatic sprinkler and the other obvious safeguards.

A New Investigation of the Fireproofing of Fabrics.

After some preliminary trials, with the assistance of the chemical engineer of the Inspection Department of the Factory Mutual Insurance Companies and conferences with the experienced scenic artists, Burrage of Chicago and Story of Boston, and after reviewing the probable effect of various solutions upon the fabrics and upon the ordinary colors used by the scenic artist with some of my personal friends who were of wide experience as chemists of textile factories and chemical works, I enlisted the ingenuity of my friend, Mr. George C. Whipple, Consulting Engineer, Director of the Mt. Prospect Laboratory in Brooklyn, and of Mr. Irving W. Fay, Professor of Chemistry in the Brooklyn Polytechnic Institute, in the hope that *starting with the theory of the successful action of ammonium phosphate*, as stated above, we could find some substance of equal value as an antipyrine that would be less likely to injure fabric or colors. Sundry theaters and scenic studios were visited by Mr. Whipple to learn the practical conditions. The bibliography of the subject was again thoroughly reviewed. Standard methods for testing the comparative efficiency were worked out, and tests were made with substantially all of the substances that had been recommended by good authorities.

Nothing was found better than, or so efficient as, the phosphate of ammonia, known to be efficient for the past fifty years. *Nothing was found that would prevent the instant burning with a rush of flame when the test was made with a strong blaze on closely hung sheets of canvas*, but many substances were found

that would make gauze and canvas proof against ignition by a match, flame, gas jet, a cigarette or an electric spark.

Some Tests of Effect of Fire-proofing Solutions upon Colors.

In my Boston tests the results of sundry solutions, after from one month to one and one-half years' time, upon canvas sized and painted in the ordinary way with the ordinary colors of the scenic artist's palette, was as given in Tables 4 and 5 (pages 127 and 128), which are representative selections from eight large test sheets:

These eight samples were prepared in the studio of Mr. Story, a well-known scenic artist of Boston, under the supervision of Mr. L. K. Davis, chemical engineer of our Inspection Department. Sheets of ordinary linen scenery canvas and also sheets of wide cotton such as used for scene painting, each about seven feet square, were painted with broad flat stripes of the colors found commonly in the scene painter's palette--put on by an experienced artist in the ordinary manner. After these strips of color were thoroughly dry other broad stripes, crosswise to the first, were applied, consisting of one stripe each of the various chemical solutions which at that time were most prominently commended for fireproofing scenery. This checker-board pattern thus permitted about 180 simultaneous tests of color and chemical on each of our eight large canvas sheets, or more than 1,000 in all.

Solutions of different strength were tried on different sheets, 15 per cent. and 25 per cent. respectively, and the further experiment was made of first applying a strong solution of each chemical to the canvas before it was sized and painted. This gave much better results and far less discoloration than when the canvas was flame-proofed after it had been sized and painted.

The reason for the less discoloration plainly is that the chemical penetrates the fiber more easily before it has been sized, and that the sizing prior to the painting locks it in and puts it into less intimate contact with the pigment.

I found, in every case, that the phosphate of ammonia affected many of the colors, and that the ammonium chloride and the strong solution of "fireproofine" were very injurious.

The treated canvas when dry and shaken gave off a dust from the chemicals.

To independently verify and extend the above tests that the

TABLE 4.—EFFECT OF FLAME PROOFING SOLUTIONS UPON SCENE PAINTERS' COLORS.

These solutions each contained 15 per cent. by weight of the given salt. Each was applied cold in the same way that flame proofing solutions are applied to scenery in practical use, it being applied by a very wet broad brush to the back of the canvas. This sheet of canvas (No. 3), about 6 by 8 feet, was of ordinary Russia linen, which had been previously sized and painted by an experienced scenic artist in the ordinary way with a broad band of each color, about 6 inches wide. The stripes of chemical solutions crossed the colors at 90 degrees. All were arranged in same order as the following table. The effect upon the colors was as given in the table below.

		Ammonium Phosphate.	Sodium Tungstate.	Aluminum Sulphate.	Ammonium Chloride.	Ammonium Sulphate.	Fire Foe.	Potash Alum Saturated.	Fire Proofine.	Nat. Fire-Proof Co.'s Liquid.	Borax Saturated.
		1	2	3	4	5	6	7	8	9	10
		(Seriously injures.) Slightly changes shade of about half the colors and shows slight efflorescence in folds on darker colors.	No change except turns No. 9 orange.	No change except makes some light colors slightly muddy and shows slight efflorescence on dark colors.	No change except serious injury to orange and American vermillion.	No change except slightly bleaches the American vermillion.	No change except slightly bleaches the American vermillion.	No change except bleaches the American vermillion in spots.	(Spoils.) Bad white efflorescence over nearly all colors; would ruin any scenic effect.	No change except to very slightly bleach the American vermillion.	No change except to turn the American vermillion to orange.
1	Chrome Yellow										
2	Yellow Ochre										
3	Dutch Pink	Fades slightly.									
4	Raw Sienna										
5	Orange Mineral	Darker, muddy.			Dark, muddy.						
6	Venetian Red										
7	Burnt Sienna										
8	Rose Lake	Lighter red.									
9	American Vermillion	Turns orange.			Dark, muddy.						
10	Black										
11	Burnt Umber										
12	Raw Umber										
13	Celestial Blue										
14	Cobalt Blue										
15	Italian Blue										
16	Chrome Green	Darkens slightly.									
17	Flake White										
18	Whiting										

TABLE 5.—EFFECT OF FLAME PROOFING SOLUTIONS UPON SCENE PAINTERS' COLORS.

These Solutions Contained Each 25 per cent. by Weight of the Given Salt.

	On Ordinary Linen Canvas.					On Heavy Cotton Sheeting.				
	Ammonium Phosphate. A	Ammonium Sulphate. B	Ammonium Chloride. C	Aluminum Sulphate. D	Sodium Tungstate. E	Ammonium Phosphate. A	Ammonium Sulphate. B	Ammonium Chloride. C	Aluminum Sulphate. D	Sodium Tungstate. E
	(Spoiled).—Slightly changes nearly all colors. Where canvas has been rolled shows white cracks or streaks of efflorescence.	Very slight changes. Slightly bleaches Amer. Vermillion. Slightly darkens Chrome Green and Cobalt Blue.	(Spoiled).—Spoils Orange Mineral and Amer. Vermillion. Very slightly darkens Chrome Green.	Very slight change except some efflorescence and makes the blues muddy.	No change in color except bleaches Amer. Vermillion and makes Rose Lake dark and muddy, but gives White efflorescence, cracks. Nos. 13 to 16. Also No. 6. Also turns the Whiting yellow.	Injures a majority of colors, making them muddy.	Turns a majority of colors darker.	Spoils nearly all the colors, making them much darker.	Very slight change except darker blotches on 8, 11, 12 and 13.	Slight changes except in Rose Lake and Amer. Vermillion.
1. Chrome Yellow.	Dirty gray.					Dirty gray.	Slightly darker.	Dark, muddy.		
2. Yellow Ochre..	Very slightly darker.					Slightly darker.	Darker.			
3. Dutch Pink....	Whitens slightly.					Slightly darker.	Darker.			
4. Raw Sienna....	Slightly muddy.					Slightly darker.	Muddy.			
5. Orange Mineral.	Very much darker.		Turns Dark Brown.			Very much darker, muddy.	Very muddy.	Turns dark brown.		
6. Venetian Red..										
7. Burnt Sienna...										
8. Rose Lake.....	Efflorescence.						Darker, muddy.			Much darker.
9. Am. Vermillion	Turns Orange.		Turns Dark, muddy Red.			Darker red and blotchy.	Slightly orange.	Turns dark red.		Turns bright orange.
10. Black..........										
11. Burnt Umber...										
12. Raw Umber....										
13. Celestial Blue..	Cracks White.					Slightly. darker.				
14. Cobalt Blue....	White Cracks.					Muddy blotches.	Much darker.			
15. Italian Blue....						Muddy blotches.	Slightly darker.			
16. Chrome Green..	Slightly darker.		Very slightly darkens.				Darker in blotches.			
17. Flake White....										
18. Whiting........	Turns Yellow.						Stains yellow slightly.			Yellows.

chemical engineer of our Inspection Department had made in a Boston scenic studio, Messrs. Whipple and Fay tested about thirty-five colors and shades by painting these colors in stripes on a sheet of canvas, and there crossing them with stripes of the various fire-proofing solutions. In the brief tests by Whipple and Fay no colors were found affected by the solutions commonly used save the cobalt blues and the delicate violets, thus differing somewhat from the results of my previous tests at Boston in which I found many of the standard scene painters' colors affected by phosphate of ammonia and ammonium chloride to the extent of changing the shade or tint, and in extreme cases destroying the color, and had found that samples of painted scenery treated with a trade preparation called "fireproofine" became marred by a dusty white efflorescence.

Possibly this efflorescence is to some extent a matter of manipulation, and the decision about injury to colors, as suggested regarding the promotion of mildew, had best be made after we are all possessed of the result of a few years' experience with the present legal requirements for fire-proofing scenery in practical use and with solutions of the strength actually applied. What little I have seen of men at work on fireproofing scenery leads me to fear that in order to avoid discoloration and efflorescence, the solutions will be put on too weak for the best flame-proofing, and that after the Iroquois is a little further in the past, most of the scenery will no longer be treated for flameproofing.

This effect of antipyrine chemicals upon colors is a question for the chemist rather than the engineer, and it is quite possible that a full range of the necessary colors could be worked out from pigments that would not be changed by the flame-proofing liquids, particularly if the antipyrine chemical be applied to the new canvas before sizing.

The Whipple and Fay Investigations on "Fireproofing" Scenery.

Mr. Whipple and Dr. Fay gave much time to testing the relative efficiency of various solutions and to developing standard methods of test, by which the relative efficiency of one fire retardant solution could be compared with another, and their work is so complete and instructive that I regret I can present here only a summary of it.

The following brief outline will show its general scope.

The results may be summed up as follows:

(1) Phosphate of ammonia was found the most efficient antipyrine.

(2) Tungstate of soda, so often found recommended in the text-books, was found to possess almost no value.

(3) The various proprietary solutions when analyzed were found to be all based on one or another of the ammonium salts, commonly the phosphate, but frequently the cheaper sulphate substituted in whole or in part.

(4) Linen canvas or cotton cloth, fireproofed in the best manner possible by any of these solutions, could be quickly burned to total destruction if a sheet were rolled in a loose coil with the axis vertical and a space of perhaps one-half an inch between the folds, and a fire then lighted with a small wad of excelsior at the bottom of the roll; this method of test serving to confine the radiant heat and the gases distilled from the fiber. This was of special interest since strips of the same cloth tested in the manner that has satisfied previous experimenters—by holding the strips of treated cloth vertically over an ordinary Bunsen flame—could not be ignited and appeared almost perfectly flameproof.

(5) The most efficient part in the fireproofing of fibers was found performed by the covering of the fiber with a non-volatile liquid that excluded the oxygen. Phosphoric acid proved better for this purpose than any other substance tested, but obviously could not be applied alone because of its corrosive action on fibers and colors.

(6) The ammonium in combination with it in phosphate of ammonia was found of value chiefly in locking up the corrosive qualities of the phosphoric acid until released by the heat of the fire, and thus giving a comparatively harmless compound for application to color and fabric.

(7) The method of application of the fireproofing solution to the canvas was found to have great influence on the degree of fire protection secured. One of the best solutions, when brushed cold over the back of old scenery, penetrated the fiber so little as to be of no value, but when applied hot was efficient. Under some conditions the linen canvas is repellent of water, as one finds on trying to dry the hands on a new crash towel. When the liquid is applied rapidly

to a vertical surface with a brush, linen cloth does not absorb it readily. Hot application of the solution adds much to its efficient penetration of the fiber. For new scenery, probably the best method is to saturate the canvas between rollers in a bath. The next best method is to mix the chemicals with the water of the glue size that the scene painter puts on before painting his picture.

(8) Tests of the tendency of the various chemicals to induce decay were made by sowing some of the treated samples with mold spores. Other tests were made by adding various per cents. of phosphate of ammonia to nutrient gelatine and to mixtures of the glue size, and incubating these for tests of bacterial growth.

(9) The effect of the solutions on the colors ordinarily used by the scenic artist was not found bad, except in case of some of the more delicate blues and greens, but a greater length of time would be necessary before positive statements about this can be made.

(10) When canvas that has been flameproofed is actually burned as it may be under practical conditions, it gives off fumes that may be even more dense and suffocating than those from the untreated canvas.

Proof that the Kind of Paint Used on Scenery Makes it Less Readily Combustible.

At the beginning, Messrs. Whipple and Fay made tests of the comparative combustibility of old painted scenery canvas with new unpainted canvas by taking strips all of the same size, thirty inches high by three inches wide, and burning them while hanging vertically from a nail, in a box that shielded them from cross drafts of air. The specimens were all lighted at the bottom and all under similar conditions.

TABLE 6.

Specimen No.	Original Weight of Sample Grains.	Time of Burning, Seconds.	Weight of Pigment Compared with Unpainted Canvas.
0	11.0	30	40 %
1	16.1	35	48 %
2	18.7	42	61 %
3	24.1	58	121 %

In No. 3 the paint was heavier than the canvas.

It will be noted the retardation of the flame was proportional to the amount of paint. The flames reached a maximum height of one and five-tenths feet.

Order of Experimenting on Effect of Various Antipyrines.

The general course of the subsequent experimenting followed by Whipple and Fay, stated briefly, ran as follows:

As a preliminary experiment, strips of cotton cheese cloth were dipped in various saturated solutions, and after drying were held in a Bunsen flame. The substances were thus quickly proved as to relative efficiency as follows: the ammonium phosphate in saturated solution proving most efficient of any.

Full notes, which I will not take space to reproduce here, were made of the behavior of each sample as a guide to further tests.

Relatively Poor Results.

Common salt.
Boric acid.
Borax.
Borax ½ and sodium sulphate, ½.
Sodium phosphate.
Sodium sulphate.
Sodium tungstate, 15% sol.
Bicarbonate of soda.
Ammonium sulphate, half sat. sol.
Ammonium phosphate ½, and sodium sulphate ½.
Aluminum sulphate.
Potash alum of various strength of sol.
"Paris Theater solution."
"Subrath's Formula."

Relatively Fair Results.

Ammonium chloride.
Martin & Tessier's formula.

Relatively Good Results.

Ammonium phosphate.
Phosphoric acid.
Borax ½ and ammonium sulphate ½.
Ammonium sulphate.
Calcium chloride, 25% sol.
"New Paris solution."

Next, explanation was sought of the reason for the behavior of the various fireproofing compounds.

Points in Theory of Flameproofing Established by the Whipple and Fay Tests.

(1) The influence of the water of crystallization in retarding ignition was studied. It was found that although different samples of cloth treated respectively with alum, borax and sodium tungstate and each loaded with all it could carry, the large amount of water of crystallization in these salts did not make them efficient fire retardants. The subject was studied further by selecting two salts, both compounds of the same phosphoric acid, but one possessing twelve molecules of crystal

water, or over sixty per cent., while the other possessed none; sodium phosphate and ammonium phosphate being chosen. The chemical reactions were studied through the successive stages, and the relative effect judged by weighing the amount of char left after ignition of the treated cloth. *It became plain that water of crystallization played a much less important part than the fluid, varnish-like residuum.*

(2) Tests were then made for learning of the influence of the ammonia given off from the phosphate of ammonia when heated by comparing the effects of potash alum and ammonia alum. The ammonia alum proved somewhat the better, indicating that the evolution of ammonia had some small value.

(3) Tests were made to learn of the efficiency of the phosphoric acid left from heating the phosphate of ammonia by starting with canvas treated with phosphoric acid. The phosphoric acid proved nearly as efficient a fire retardant as the phosphate of ammonia. *The chief value of the ammonia in the phosphate of ammonia appeared to be the rendering of the phosphoric acid less harmful to canvas and colors.*

(4) A study was next made of the absorption of heat by the volatilization and decomposition of the fireproofing salts, and it was, for example, made apparent that the number of thermal units absorbed in driving out the combined water from a given weight of ammonium chloride was nearly four times as great as for an equal weight of sodium phosphate, and this helps make clear why ammonium chloride has flameproofing qualities of some value, while the sodium phosphate is comparatively worthless for this purpose.

(5) A study was then made of the combustible quality of the gases distilled off when canvas that had been treated by various flameproofing compounds was ignited, in order to learn if inert gases derived from the chemicals used for flameproofing diluted the combustible gases from the cellulose, to the point where the combined gases would not ignite. For this purpose little rolls of linen untreated, and treated by various chemicals, were heated to destruction separately in glass ignition tubes, five-eighth of an inch diameter x 6 inches long, placed with the end in a muffle, heating the muffle by gas to a temperature which, judging by the color, was from one thousand degrees to twelve hundred

degrees Centigrade. This temperature, as shown by the color, was maintained nearly constant all through these ignition tests.

These test rolls were two and one-half inches long and lay only within the uniformly heated zone at the bottom of the test tube. The distilled gas issuing from the end of the test tube was ignited. The progress of the charring of the canvas could be observed through the glass tube. The relative amounts of tarry matters condensed at the cooler, outer portion of the tubes was also compared.

It was found that canvas, flameproofed so that a strip of this canvas could not be made to ignite from a Bunsen flame, would, when tested in the ignition tube, not give off ignitible gases from the tube. The rapidity and simplicity of the ignition tube test were found such as to commend it.

Therefore, series of tests with the whole line of known efficient fire retardant compounds was made in this manner, and full notes of their behavior kept.

As a result of the tests thus far, it was concluded in brief:

(*a*) That inert chemical substances can exert but very slight fire-retarding action.

(*b*) The fire-retarding action of salts which depend for fire-retardant quality only upon their water of crystallization, like potash, alum, sodium phosphate and borax, is slight and unimportant, although somewhat superior to that of inert substances.

(*c*) Fire retardants of the class which suffers chemical decomposition under heating are decidedly more efficient than those which depend on the driving off of water of crystallization, but still far less efficient than the class that follows.

(*d*) *The most efficient salts are those which on decomposing leave behind a non-volatile residue which is fluid at the temperature of the burning canvas, and covers the charring fabric with a thin glaze which prevents further access of air*, and of this type, phosphate of ammonium was found to be the best.

Analyses of Sundry Proprietary Fireproofing Solutions in Use in 1904 *to Meet the Recent Requirements of the New York Building Law.*

The following table gives the result of chemical analysis of the most prominent fireproofing solutions found at that time on sale

in the New York market for the purpose of fireproofing scenery:

	Grams per 100 Cubic Centimeters.							
	Total Residue.	Ammonium Sulphide.	Ammonium Phosphate.	Ammonium Chloride.	Sodium Sulphate.	Borax (Crystals).	Boric Acid.	Sodium Tungstate.
1. Fireproofine	....	6.3	1.5	....	....	2.5	2.3*	....
2. H. S. Fireproofing Solution	16.5	....	... .	12.6	....	....	....	....
3. Electric	24.5	....	17.0	....	9.7	....	....	....
4. H. W. Johns' Compound	16.7	10.7	....	....	....	5.3	...	....
5. No flame	15.5	8.4	5.0	....	....	....	...	....
6. Blenio Solution	20.7	3.7	15.6	....	....	1.4	....	....
7. Salamanderine	17.4	8.1	....	5.8	....	3.6	....	....
8. Antipyros Klugiana	23.9	....	18.5	....	2.0	1.0	....	2.5
9. Van Ripper Solution	....	present	....	....	....	....	....	....
10. Lamb & Finlay's prepared canvas	....	present	present	....	....	....	....	....

* Combined as glyceride, 3.9 grams per 100 C.C.

Tests were made on scenery canvas that had been treated, *the fabric being thoroughly impregnated* by soaking and wringing out or by brushing on both sides of the heavy canvas, with each of the foregoing, both in the Bunsen flame and in the glass tube in the furnace; all of them were found to be fairly efficient, Nos. 9 and 10 being perhaps the least so. It should be noted that *the cloth tested was more thoroughly impregnated than old scenery will be when brushed over on the back at a single application.* Those solutions containing the larger amounts of ammonium phosphate were found the most efficient. The only apparent advantage of the chloride or sulphate of ammonium is the fact that it costs only half as much as the phosphate; it is less efficient.

Sodium sulphate, boric and boric acid are present in some of the solutions. These were found to contribute relatively little to the flame-resisting power, and the sodium tungstate came to be regarded by these chemists as worthless for this purpose.

U. S. Patents on Fireproofing Solutions.

Previous to the investigations made for me by Messrs. Whipple and Fay I had procured from the U. S. Patent Office a complete file of the patents issued during a period of about 30 years, for the purpose of studying them for suggestions as to chemicals or processes to be used. I found in them nothing of particular interest. The compounds in most cases were made up by mixing one and another of the salts, alum, phosphate of ammonia, borax, sulphate of ammonia, etc., that have been in common use and recommended over and over again for 50 years,

the novelty consisting in the precise formula for proportioning the mixture and in the selection of ingredients. Alwin Nieske, of Dresden, Germany, however, went somewhat outside the beaten path in patenting in 1901 molybdate of sodium in a 10 per cent. solution for application to fabrics for fireproofing and preserving them. In general, fabrics other than theatrical scenery appear to have been in the mind of these patentees, and the number of patents for fireproofing textile fabrics is not nearly so numerous as for the fireproofing of wood.

Difficulty of Proper Application of Fireproofing Solutions.

In the foregoing tests the effort had been to test the efficiency of the solutions, it being assumed they would all be most thoroughly applied.

The method of application of fire-proofing solutions was next made an object of study by Whipple and Fay. Samples of old scenery were subjected to treatment by the various more efficient solutions in different ways, and finally in order to produce uniform results and ensure the uniform distribution, *application was made by immersion in a bath* containing submerged rollers, while dipping, followed by a wringer with rubber rollers was used as an alternative method.

It became plain that the method of application and the thoroughness with which the solution was absorbed had much to do with efficiency. In order to completely saturate the fibers, many dips and wringings were found necessary. The glue size of the unpainted back of scenery canvas prevents to a considerable degree the rapid penetration and absorption of the liquids applied. Under rapid application with a brush, the best of the solutions may fail to render the canvas flameproof, particularly if applied cold.

No inspector can tell from the appearance of one of these large sheets of canvas whether the solution has been properly applied all over its surface, and probably all that inspection will ordinarily amount to in practice will be equivalent to what would be shown by the touching of a lighted match to the edge of the canvas sheet.

As to the permanence of the residue left in the canvas, it was noted that when cloth that had been treated by one of the best

of the fireproofing solutions was shaken and brushed, the white powder could be shaken off in the form of dust, and that more was removed by brushing. This indicates that although a freshly treated canvas may be well flameproofed, it may lose this quality to a noteworthy extent after the rough usage which scenery received on the stage and on the road. It would be interesting to follow this matter further by tests of pieces taken from the margins of scenery that had been treated, and then had one or two years of travel and use.

Mildew.

It has been claimed that the application of fireproofing solutions weakens the canvas. If true, it is important to know whether this comes from slow chemical action or from the rotting of fiber due to bacterial action or mold, since in the latter case a germicide could perhaps be incorporated in the solution.

Tests of effect of certain fireproofing compounds in promoting mildew and mold were made; first, by adding varying percentages of ammonium phosphate to nutrient gelatine which was then exposed and incubated by methods common in bacteriological work, and, secondly, by seeding the worst treated canvas with mold spores. Time was lacking to carry these tests to the desired length, but so far as they went it was found that the glue used to size the canvas is probably a more potent promoter of mildew than the salts employed for fireproofing. Concentrated applications of the fireproofing salts will doubtless retard these organic growths, while dilute applications of some of the salts, phosphate of ammonia, for example, will very likely stimulate mold and bacterial decomposition, particularly if hygroscopic. Time did not permit the working out of experiments to find a suitable germicide for addition to the solutions.

The Fireproofing of Wood.

Since the pine frame work of the set pieces and wings present a greater quantity of fuel than the canvas itself, it would be desirable to flameproof this wood. A simple brushing over with phosphate of ammonia or other chemical solutions is found inefficient.

Various processes for making wood fireproof have long been known and have been used on wood for interior finish and trim

of fireproof buildings, more here in New York City than anywhere else, because of certain favoring clauses in its building laws.

The various tests made by Professor Norton of the Massachusetts Institute of Technology * and others have shown that, although the wood, after treatment, is much less readily ignited from a small blaze, as from a match or an electric spark, *no real fireproofing results.* Previous tests have covered this matter so thoroughly, and have shown the loss of strength and tendency to gather moisture and other objectionable qualities that follow treatment, that I gave little attention to testing this matter further, but rested mainly on the tests of previous experimenters. I obtained sundry specimens of wood that had been fireproofed in the commercial way from two prominent shipyards that had war vessels under construction and I made a few simple tests.

Fireproof wood was at one time much used on the war vessels of the Navy, but has been almost wholly abandoned by reason of its gathering moisture badly and the lessening of strength and the increased difficulty of working it.

The frames of scenery must be particularly light and strong, and the wood must possess its maximum strength, and should not be liable to warp. I do not find that "fireproof" wood has ever been used practically for this purpose at any theater, in this country or abroad, notwithstanding the activity of its promoters. I soon concluded that in the present state of the art it was too much to expect that the wood flameproofed by any of the ordinary commercial processes could come into general use for battens, frames, profiles, etc., of stage scenery.

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* See "Report on Fireproof Wood," so-called, by Prof. C. L. Norton, August, 1902.

Professor Norton summed up the results of his tests on samples of wood "fireproofed" by three of the more prominent commercial processes as follows:

"Fireproofed wood is almost identical with untreated wood in the following particulars:

"It smokes at about the same temperature.

"It can be ignited at about the same temperature.

"It will continue to burn in many cases.

"It is a good fuel.

"It makes a very hot fire."

The ordinary method of test of little samples in the flame of a laboratory lamp tends to greatly exaggerate the extent of protection against fire gained. A better test is to make a small long vertical box of the wood, open at top and bottom, and let this serve as a chimney for a small fire kindled inside at the bottom.

"Fireproof" Paints.

"Fireproof paints" are sometimes required by law to be applied to wood-work about the stage. The underwriters' laboratory at Chicago had a short time previously made an extensive series of tests of all of the prominent ones in the market. The unpublished records were placed at my service. These tests had shown that none of these paints had any noteworthy value in flame-proofing wood, but for confirmation I requested Messrs. Whipple and Fay to make tests of a few of those most prominent in the market. They purchased commercial samples and made chemical analyses of several; each was found to be mainly a sort of whitewash consisting of slaked lime, finely pulverized asbestos, with also a little alum, gypsum and glue. The paint adhered well when applied to canvas, but was quickly proved by test to have almost no flame-proofing quality whatever.

It is difficult or impossible, on precise scientific grounds, to see how these paints can have any noteworthy value against anything but a very small momentary blaze, like that of a match or spark.

None of these paints were found to penetrate below the surface of the wood as phosphate of ammonia, for example, penetrates into the fiber of cotton or linen cloth.

Obviously, so thin a film can have only exceedingly small effect as a non-conductor of heat. Radiation or contact must char the wood beneath almost as quickly as if the paint were not there. The destructive distillation will give off gas which will push out, blister, and peel off the paint, and this gas will burn.

In the "asbestos paints," the pulverized asbestos, glued into a thin crust less than $\frac{1}{100}$ inch thick, can obviously be of no more fire retardant value than so much carbonate of lime or clay. The special value of the asbestos in paints is chiefly as a name to conjure with in attracting purchasers.

From all these tests a common lime whitewash appears to be as efficient a fireproof paint as anything yet found in the market.

Development of Standard Methods of Test of Flameproofed Fabrics.

Finally, much attention was given to devising a standard method for testing the relative efficiency of various chemicals

used for the flameproofing of scenery canvas. It has already been explained that no fireproofing of cloth is effective against severe heat, but it was plain, from the preliminary trials, that some of the solutions were much better than others in protection against a little blaze like that from a match, a cigarette or an electric spark.

Since all samples, however flameproofed, were destroyed by a severe test, all of these tests, of necessity, had to be merely comparative, and canvas treated with a saturated solution of phosphate of ammonia thoroughly worked into the fiber was adopted as the standard of comparison.

The "Stovepipe" Test.

In the effort to more nearly follow practical conditions, one set of tests was developed on the line of my earlier stovepipe experiment by burning fireproofed canvas within a piece of five-inch stovepipe two feet long lined with asbestos, as shown in Figs. 10 and 10a. Six strips of the canvas, thoroughly treated with the different solutions, were placed three-fourths of an inch apart and ignited by burning one ounce of excelsior. *In every case the canvas burned completely to ash in from three-fourths of a minute to one and one-half minutes, with flames which often extended two feet above the top of the stovepipe.* Tests in the stovepipe apparatus on the efficiency of different flameproofing chemicals were made comparable by taking the same quantity of canvas in each and by lighting the fire with the same quantity of combustible.

In the first efforts to standardize the "stovepipe test," it was found, after considerable experimenting, that by using a piece of the untreated canvas eight inches high by three inches wide for a kindling piece, and pinning this to the bottom of a strip of the flameproofed canvas sixteen inches long hung from the top, the flames from the kindling piece would barely reach to the top of the pipe, and as ammonium phosphate had proved the most efficient of the chemicals used in previous tests, the behavior of a strip of canvas thoroughly impregnated with this was taken as the standard for comparison.

The height of flame did not prove a good basis for comparisons because of the varying weights and thickness of canvas and the varying amounts of glue and paint applied. The amount of char

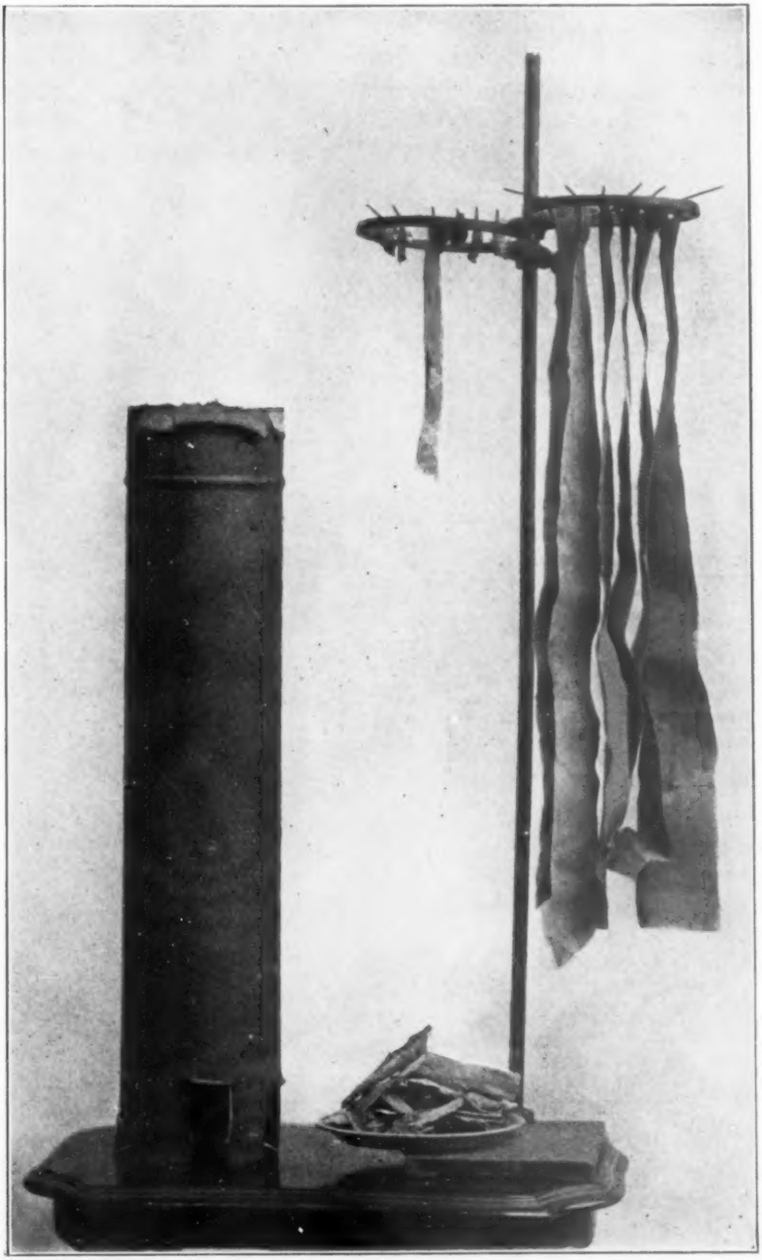
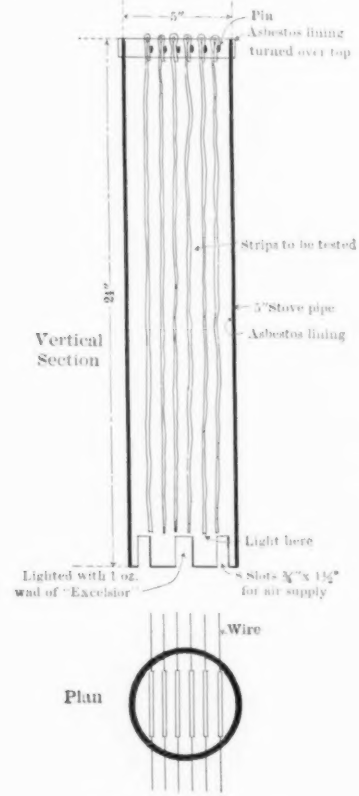


FIG. 10.—THE "STOVEPIPE" TEST OF SCENERY CANVAS.

produced was found the best basis of comparison. A strip thus prepared and lighted by the strip of untreated canvas, as above described, is for a short time bathed in flame from the burning of the strip below. A single strip thus tested alone, when taken out of pipe, is found with its lower end blackened and



"Stove pipe" for testing
"Fire proof" Canvas

FIG. 10A.

charred up for about half the length, and the upper end white and unscorched, but on *placing several flame-proofed strips side by side* in the stovepipe, *all were consumed.*

In the following tests (see Tables 7 and 8) six strips of canvas thoroughly treated were placed side by side, three-fourths of an inch apart, and ignited by burning one ounce of excelsior.

In each case all was burned to ash in from forty-five seconds to ninety seconds with flames which often extended two feet above the top of the pipe.

Fig. 10 shows a photograph of the apparatus and Fig. 10A a sectional drawing of it. The little material left hanging upon the ring at the left of the retort stand, with the charred material on the tray beneath, shows what remained from one of these tests. On the ring at the right are shown the strips as prepared for insertion in the pipe.

TABLE 7.—STOVEPIPE TEST ON NEW UNPAINTED FIREPROOF CANVAS.

Solution Used.	Weight of Strip of Canvas before Testing (Grams).	Weight of Ash Remaining (Grams).	Weight of Substance Consumed (Grams).	Per Cent of Substance Burned.
Borax	52	0.7	51.3	99
Ammonium chloride	62	0.5	61 5	99
Sodium phosphate	58	1.5	56 5	97
Aluminium sulphate	62	3.0	59.0	95
Ammonium phosphate	72	23.0	49.0	68

It will be noted that the ammonium phosphate gave the best result.

TABLE 8.—STOVEPIPE TEST ON OLD FIREPROOFED SCENERY.

Solution Used.	Weight of Strip of Canvas before Testing (Grams).	Weight of Ash Remaining (Grams).	Weight of Substance Consumed (Grams).	Per Cent. of Substance Burned.
* a. Chicago Solution	78	18	60	77
† b. Ammonium Sulphate	77	21	56	73
‡ c. Ammonium Sulphate	16	5	11	69
d. Ammonium Phosphate	102	40	62	61
(1) Fire Proofine	136	64	72	63
(2) H S. Compound	91	23	63	69
(3) Electric	101	35	66	65
(4) H. W. Johns'	91	28	63	69
(5) No Flame	93	36	57	61
(6) Blenio	97	30	67	69
(7) Salamanderine	94	27	67	71

Here, with the old painted scenery, as in the series just above with new canvas, nothing was found better than the ammonium phosphate.

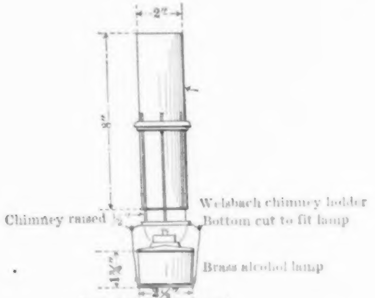
This test, although so simple, is so severe that the specimens show little difference in quality of the fireproofing.

The per cents. in this table do not strictly represent the fire-retarding action, since the per cent. is figured on the original weight, including the incombustible mineral pigment.

* Ammonium phosphate, ammonium sulphate, ammonium chloride, borax and boric acid.
† On old scenery like the others, except c.
‡ On gauze.

Lamp Tests of Flame-proofed Scenery.

Although no solution found would protect the canvas so that it could withstand a severe test, it appeared desirable to devise some simple portable standard means of comparing the efficiency of various trade solutions with that of the standard phosphate of ammonia thoroughly worked into the fabric—something that an inspector of the city building department could use on his round for finding out if the law which requires flame-proofing of scenery canvas has been complied with, if he desired something more like apparatus than a box of matches or a plumber's gasoline



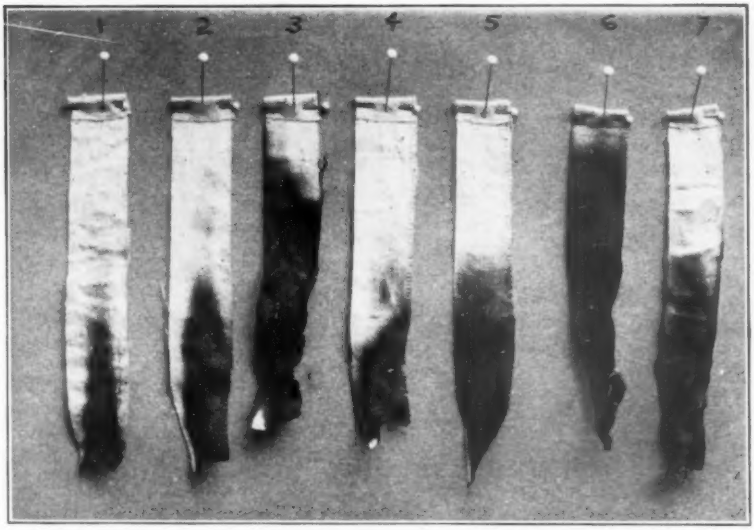
Portable lamp as used by Whipple and Fay for testing comparative efficiency of the Flame-proofing of Canvas by various chemicals

FIG. 11.

torch, or something that would permit a more definite record of the degree of resistance.

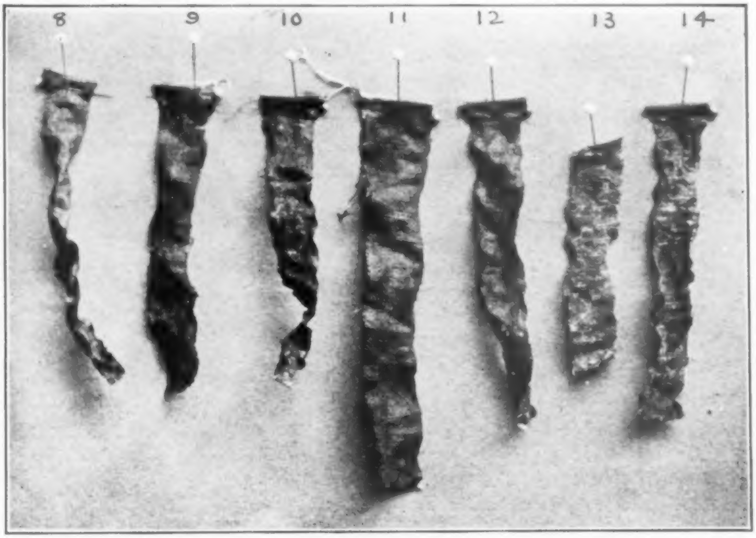
The testing lamp finally adopted by Messrs. Whipple and Fay is shown in Fig. 11.

The apparatus consists of a common alcohol lamp two inches high, two and one-half inches in diameter, fitted with a Welsbach chimney holder. The chimney served to protect the flame from side drafts, and thus to some extent prevented the dissipation of the gases. The chimney also served to support the sample and keep it in a central position over the flame. The chimney was raised half an inch above its seat in order to allow air to enter freely. The strips to be tested by the lamp were cut eight inches long and one inch wide. Each strip was folded over one-half inch at the top, so as to allow a slender wire to be passed



1. Ammonium Phosphate.
2. Antipros Klugiana.
3. Subrath's Formula.
4. New Paris Solution.
5. Martin & Tessier's Solution.
6. Salamanderine.
7. "No Flame."

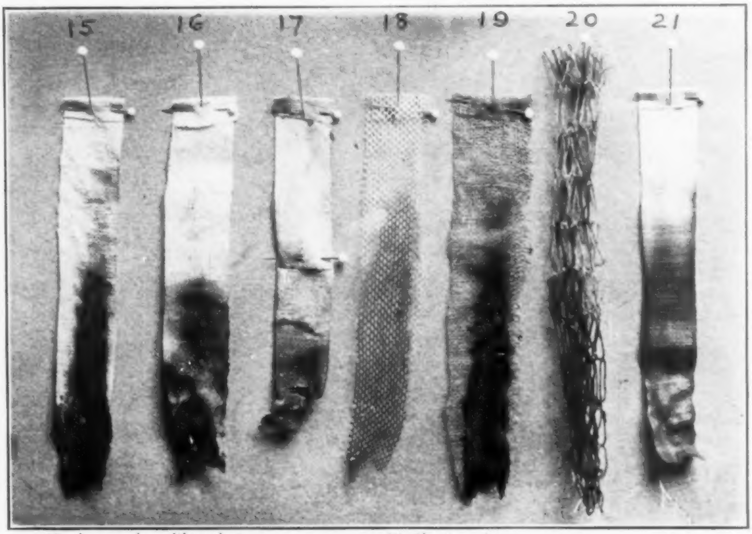
FIG. 12.



8. "Electric Fireproofing Solution."
9. Fireproofine.
10. Sodium Tungstate.
11. Boric Acid.
12. Ammonia Alum.
13. Som Phosphate.
14. Potashdiu Alum.

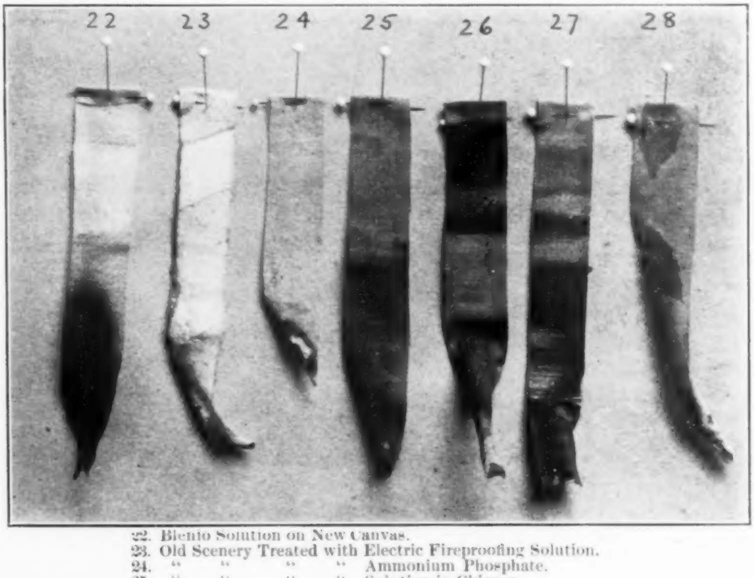
FIG. 12A.

FLAME-PROOFED CANVAS STRIPS ONE INCH WIDE AFTER TEST OF ONE MINUTE IN ALCOHOL LAMP FLAME TWO INCHES HIGH, SHOWING COMPARATIVE EFFICIENCY OF VARIOUS "FIREPROOFING" CHEMICALS FOR PREVENTING IGNITION BY A PETTY FLAME.



15. Ammonium Phosphate.
16. Ammonium Sulphate.
17. Ammonium Chloride.
18. Gauze.
19. Scrim.
20. Netting.
(18–20: Ammonium Phosphate.)
21. Canvas Fireproofed in Manufacture.

FIG. 12B.



22. Blenio Solution on New Canvas.
23. Old Scenery Treated with Electric Fireproofing Solution.
24. " " " " Ammonium Phosphate.
25. " " " " Solution in Chicago.
26. " " " " Blenio Solution.
27. " " " " Salamanderine.
28. " " " " "No Flame."

FIG. 12C.

FLAME-PROOFED CANVAS STRIPS ONE INCH WIDE AFTER TEST OF ONE MINUTE IN ALCOHOL LAMP FLAME TWO INCHES HIGH, SHOWING COMPARATIVE EFFICIENCY OF VARIOUS "FIREPROOFING" CHEMICALS FOR PREVENTING IGNITION BY A PETTY FLAME.

through it for support. This wire, resting across the top of the chimney, supported the strip of canvas so that its lower end hung into the top of the flame for one-half an inch. The alcohol flame was kept at a constant height of two inches.

The tests of canvas made in this way corresponded in results to those made in the stovepipe, with the advantage of speed in testing and of being able to see through the glass chimney what was actually taking place. The time of a single test was one minute. The accompanying photographs, Figs. 12 to 12c, show the result of a lamp test upon canvas that had been *thoroughly impregnated* with various solutions and dried. It is doubtful if in practice scenery canvas would be so carefully impregnated with the solution, and doubtful if all the solutions would be made as strong.

This test is useful, after all, mainly for the purpose of comparing the efficiency of one method of flame-proofing treatment with another with greater precision than by the common rough test of holding a small strip of the fabric in a gas flame, and it should always be kept in mind that canvas which shows little effect of burning in this test can easily be burned to total destruction in the stovepipe test, and that *canvas which appears well fireproofed by these little single-strip, lamp-flame tests would doubtless burn with a rush of flame and suffocating smoke in a theater fire.*

For the practical purpose of seeing if the law has been complied with, and the scenery flame-proofed enough so a match, or gas jet, or electric spark will not ignite it, a simple test with a plumber's torch, or even with burning matches applied at the frayed edges and seams, in the hand of a thoroughgoing inspector will serve all practical purposes.

Dry Powder Fire Extinguishers.

On and about the stage of the Iroquois Theater were several tubes of Kilfyre, so-called, one of the numerous "dry powder fire extinguishers," in a long red tube, that have been so vigorously pushed into notice by enterprising salesmen during the last few years. One of the men on the stage promptly and courageously tried to extinguish the fire with this powder. The burning scenery and the fireman were not in the best positions for an extinguisher of this kind to make its best showing, and, of course, he accomplished nothing whatever, except the loss of valuable time. The fact that such unreliable material was relied on there, and is to-day hung up in public places where it gives a false sense of security, prompts me to devote some little time to this subject.

The chief reason why these long tin tubes of dry powder have become popular is that they can be manufactured for about ten cents each, and that they retail as high as $3.00 each.

They are nearly all composed of common bicarbonate of soda (or cooking soda), frequently disguised by the admixture of a little cheap coloring matter like Venetian red, and prevented from caking by the addition of starch.

I procured a set of the U. S. patents on fire extinguishing compounds of this class and studied them for suggestions as to some more potent salt than the bicarbonate of soda, without success. In the several patents the claim for novelty generally rests on the proportion of the mixture with venetian red, yellow ochre, fullers earth, starch, etc., added to the bicarbonate of soda to prevent caking.

The party who recommended and sold these tubes of Kilfyre to the Iroquois was, I am assured, an honest man who fully believed in their efficiency, and in an effort to save others from like mistakes, I have had samples of everything of this kind that I could find in the Chicago market, the Boston market and the New York market purchased in the ordinary channels of trade by different parties, and the respective groups of samples analyzed by three different chemists, in order to fortify myself against the possibility of wronging anyone through a mistake in the analysis, and have had samples sealed up and retained for further analyses should anyone question my figures.

Fig. 13 is from a group of these extinguishers from which samples were taken for test.

Bicarbonate of soda or common cooking soda or kitchen saleratus is seen to be the principal ingredient in every case.

The bicarbonate of soda can be purchased in quantity for about one and three-fourths cents per pound. Each tube commonly contains two and one-half to three pounds. The cost of the tin box and its gorgeous label may be enough to bring the whole up

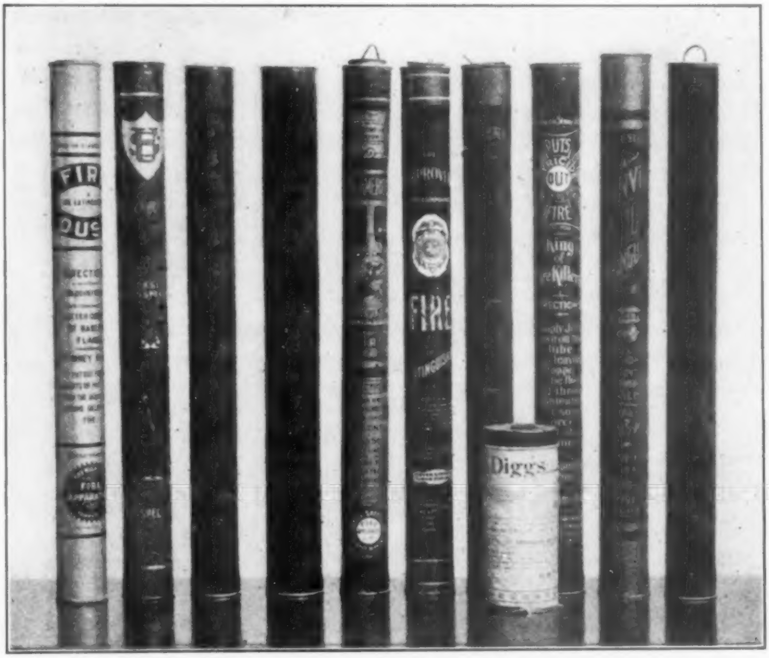


FIG. 13.—A GROUP OF DRY POWDER FIRE EXTINGUISHERS.

to ten or fifteen cents. If these are what one wants, why pay from two dollars to three dollars apiece for them? Why not buy a package of common kitchen "saleratus" at the grocer's?

I have heard remarkable stories of what they will do. Remarkable exhibitions are sometimes given under circumstances specially devised. My New York friend, the chemist, was given an exhibition by a man who poured a thin stream of benzine on the floor, lighted it and extinguished some of the powder. My friend was impressed, but did some experimenting at home and

A table of representative analyses follows:

TABLE 9.—ANALYSIS OF DRY POWDER FIRE EXTINGUISHERS.

Price per Tube.	Name of Dry Powder Fire Extinguisher.	Per Cent. of Chemicals by Weight.											Fineness. Per Cents. through Sieves.		
		Bi-Carb. Soda.	Red Ochre.	Yellow Ochre (or Iron Ore).	Fuller's Earth.	Common Salt.	Amm. Carb.	Sodium Phosphate.	Nitrate Soda.	Charcoal.	Fire Clay.	Sodium Sulphate.	200 Mesh.	100 Mesh.	80 Mesh.
	"Kilfyre"	97.5	2.5										86	100	100
$3.00	"	96.0	4.0												
3.00	"Kingfyre"	97.4	2.6										75	86	87
3.00	Fyrcide	99.5	0.5										42	65	98
	"Fire Dust"	94.8		5.2									82	99	99
3.00	"Improved"	87.6		12.4									38	98	99
3.00	Atomized	95.0		5.0									95	98	99
	Electric														
0.75	Driggs	89.5			10.5								70	92	96
3.00	Pan-American	30.6		37.9		7.4	2.3	8.1				12.9	4	54	73
3.00	Phenix	81.2		4.0							14.8		31	94	98
3.00	Phenix	82.1		17.9					2.6	0.7					
3.00	Manville	95.5		1.3									26	97	98
3.00	Manville	96.0		4.0											

Another series of analyses ran as follows:

TABLE 10.

Price per Tube.	Name of Dry Powder Fire Extingnisher.	Per Cent. of Chemicals by Weight.													
		Na_2O Soda.	CO_2 Carb. Acid.	Insol. Matter (Iron Ore).	Water in Bi-Carb. by Diff.	NaCl Common Salt.	NH_3 Ammonia.					Na_2So_4 Sod. Sulphate.	Starch.		
$2.50	Kilfyre	39.4	50.6	4.0	10.0										
3 00	Pan-American	15.3	20.3	41.1	14 1	6.7	0.4					2.0			
1.00	Eclipse	32.6	43.2	3.4	12.0								8.8		
2.00	Manville	35.0	48.0	4.0	13.0										
3.00	Phenix	30.1	41.8	17.6	10.1										

Another series as follows:

TABLE 11.

	Name.	Bicarb. Soda.	Hono. Carb. Soda.	Iron Oxide.	Insol. Dust.	Loss at Red Heat.	Amm. Carb.	Common Salt.	Sodium Sulphate.	Insol. Iron Ore.	Water, etc.	Starch.	Clay.		
	Monarch	93.7	2.4	3.8		35.7%									
	Pyricide	94.7	4.6	0.7		36.8%									
	Pan-American	34.5	4.4				1.1	6.7	2.0	41.1	10.2				
	Eclipse	37.9										8.8	3.3		
	Kilfyre	96.5			3.5										
	Swan	97.2	1.7		1.0										

found that after a little practice he could do the same with either sand or salt. We had tests made of two of them by our inspectors a few years ago and found them of doubtful value on the smallest fires, and worthless for a fire in free ventilation. They show up particularly well in a little fire kindled in an office spittoon.

No doubt, the material has some small value for a certain class of fires. Doubtless, it is wise to carry a few tubes of this on an automobile. Doubtless, in confined situations, on the apron of a cotton picker, even the bicarbonate of soda powder may sometimes do remarkably well, but *Dry Powder Fire Extinguishers should never be used to give a false sense of security about the stage of a theater.*

We do not recommend these tubes of dry powder in factory fire protection. We recommend they be thrown into the rubbish heap. Pails of water are far more reliable.

On the other hand, the "soda water fire extinguishers," consisting of a copper cylinder containing two, three or four gallons of a strong solution of bicarbonate of soda, with a bottle of acid at the top so arranged that it can be upset into the soda and water, thereupon generating a strong pressure by the evolution of carbonic acid gas, are excellent for many situations where pails would be unsightly.

Hand Grenades.

These are glass bottles, commonly of roughly spherical shape, and holding about a quart each of a liquid that it is claimed possesses marvellous fire extinguishing properties. I found many of these scattered about in some of the older theaters.

As showing what people will pay good money for in the effort to get fire protection, I was interested in the story that one of my agents, a chemist, in collecting samples, brought in about hand grenades. We had purchased examples of some of the different kinds of hand grenade, and had brought home a few samples that we found hanging in theaters and had their contents analyzed. In the case of particular interest, the salesman offered, as proof of the superior merits of his compound, the statement that a quantity of his particular make and style of hand grenade had just been purchased by the United States Government for the protection of one of the battleships. Our analysis shows the

contents to be simply water and common salt. I myself saw a hand grenade of the same appearance, bearing the same label, in the model of the battleship at the St. Louis exposition, so perhaps it is true that the United States government purchased salt water at fifty cents per quart bottle for the fire protection of battleships.

The chemists reported the following analyses in certain samples of hand grenades. As stated, most of these samples were old and not direct from the maker.

ANALYSIS OF CONTENTS OF "HAND GRENADE FIRE EXTINGUISHERS"

"Hayward" hand grenade, specific gravity of solution		1.188	
	common salt	22.3	per cent.
	other solids	0.4	" "
"Harden" hand grenade, common salt		18.5	" "
	salammoniac	6.7	" "
	Total	25.5	" "
"Babcock" hand grenade, common salt		21.2	" "
	chloride of calcium	6.5	" "
		27.7	" "

These materials are inert, and their only advantage over plain water is that they do not freeze at ordinary winter temperatures.* The hand grenades contain about one quart of water, while a 30-cent fire-pail holds ten quarts and costs less.

I have been much interested in collecting a file of all of the patents of the United States patent office for hand grenades and fire extinguishing compounds. There are many of these patents. They are interesting reading, but I judge them more curious than useful.

A favorite line of some of the patentees has been to devise a compound apparently on the theory of finding something that would burn quicker than the surrounding fuel and thus by exhausting the oxygen smother the first fire. Other patentees propose mixtures that generate sulphurous acid and ammonia gas because of their non-support of combustion, in sublime disregard of their poisonous non-breathable quality.

Several subjects remain which we have scant time to discuss.

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* For places where a non-freezing inert liquid is desired for filling fire pails probably there is nothing yet available better or cheaper than a strong solution of chloride of calcium in water. This is obtained as a by-product in the soda works of the Solvay Process Co. at Syracuse (perhaps elsewhere also), and has recently been put on the market at a low price. It is largely used for the circulating liquid in refrigerating plants.

The most important is the fire escape. I will take time only to call attention to a source of fatality that had not been foreseen until the Iroquois fire.

A Fire-Trap "Fire Escape."

In Fig. 14 the fire and smoke issuing from the door marked *F* ascended and enveloped the fire escape leading down from

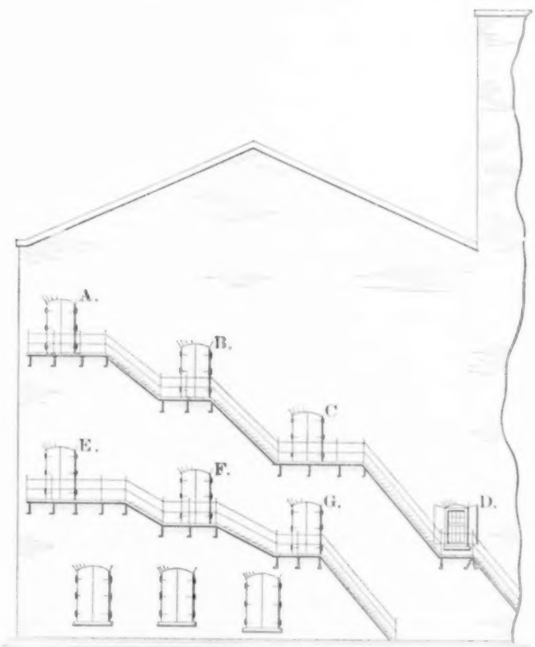


Fig. 14.—Emergency Exits in Rear of Iroquois Theater.

A Fire Trap Instead of a Fire Escape. Flames Issuing at F Cut Off Escape from A, by Enveloping Gridiron Platform at B in Flames.

the upper gallery, so that many who crowded out through the doorway and stood on the upper platform at *A* could not descend, and several in their terror jumped about 40 feet to their death on the hard ground below.

I fear that in many of the theaters similar conditions could arise to-day, and this great danger of a window or doorway

underneath, through which the flames can issue and envelop the fire escape, and thus cut off its use, should be carefully looked out for.

Philadelphia Fire Escape.

A type of fire escape has been developed under the Building Laws of Philadelphia primarily for use in factories, which is so remarkably efficient and so far ahead in safety of anything else that exists that we may wonder why it has not been copied in other cities. True, it is somewhat expensive, but the safety it gives is well worth the extra cost. The same idea can be readily applied to the fire escapes from a theater.

Two varieties of this are shown in Figs. 15 and 16; one known as the Balcony type and the other as the Tower type.

The fundamental idea is that the stairway tower is absolutely cut off from the various rooms and floors which it serves. One must go out from the room into the open air and then enter the stairway. Once within this stairway tower, he can proceed without danger to the bottom.

It is to be noted that in the Tower type (Fig. 16) the free opening in the top of the tower extends close to the bottom of the floor above, while the doorways for the same story have their tops at a much lower level. Therefore, any smoke coming from an opened door of the workroom will, as it rises, find escape to the front opening at a much higher level than the door from which it issues, and will not tend to enter the door into the stairway tower, which has its top at so much lower height than the free opening in front. The stairway is thus free from danger of flame or smoke, and presents safe outlet for workmen and safe means of access for firemen.

Escape from the Gallery.

The great lesson out of all the theater fires as to the danger to those in the gallery should not be forgotten in designing the stairways and fire escapes. The area, the total number of stairway exits, and the *total width of stairway per hundred persons should be two or three times as great for the gallery as for the other parts of the house*, and all exits should run in such a direct and obvious course, with guide curves instead of abrupt angles at changes of direction, that with a person once in them, he could

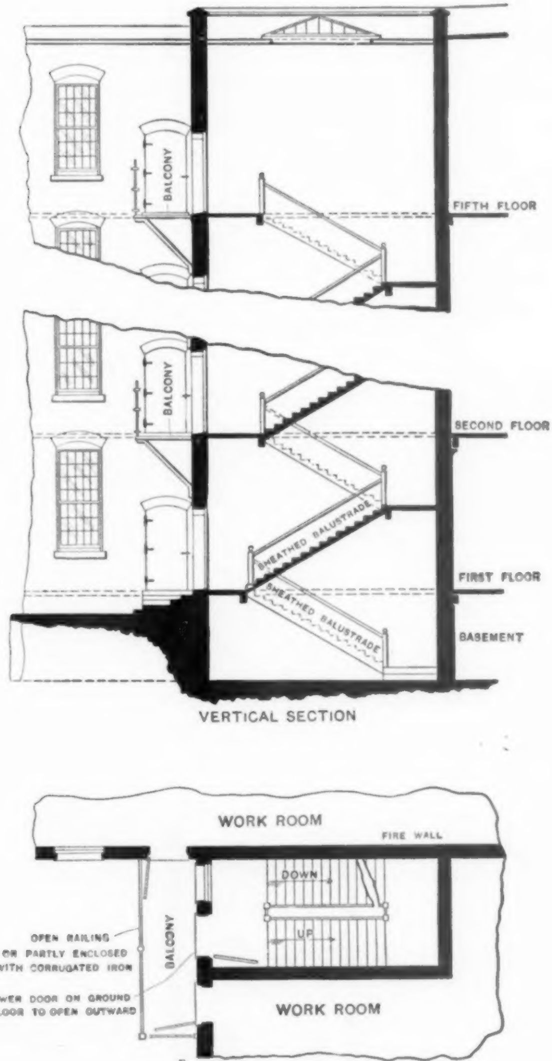


FIG. 15.—BALCONY STAIR TOWER AND FIRE ESCAPE FOR FACTORIES. PHILADELPHIA TYPE.

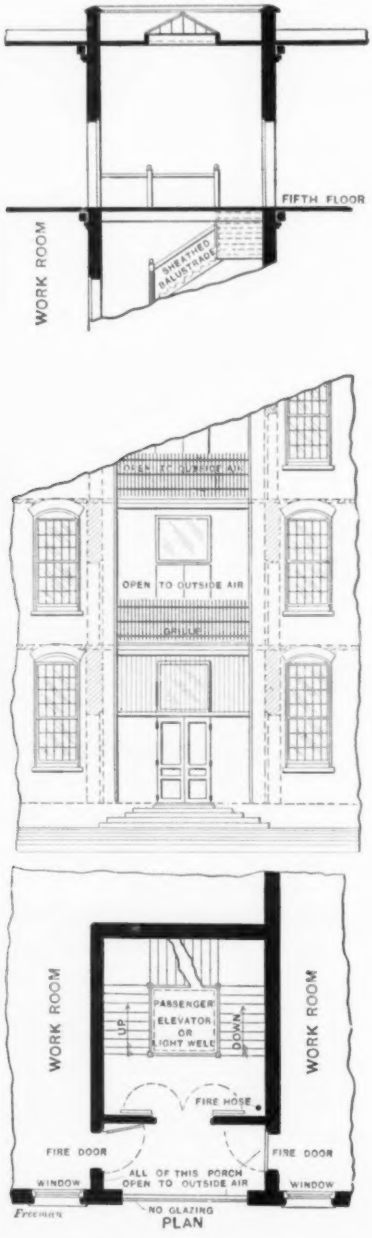


FIG. 16.—TOWER FIRE ESCAPE FOR FACTORIES.
PHILADELPHIA TYPE.

not fail to find his way to the bottom, although in total darkness. The flights of stairs should be each of the fewest steps practicable, with frequent landings on which one can steady himself, and with good, simple, continuous handrails on each side that can be followed down in darkness by sense of feeling, and a strong centre rail, continuous all the way, where wide stairs are necessary.

Width alone, as prescribed by most building laws, is not the sole consideration. The architect of the Iroquois testified that *the gallery exits of the Iroquois were of 100 per cent. greater total width than the law required.* Yet 70 per cent. of those in the Iroquois gallery perished, many at the back of the room not reaching the exits, and some in their seats.

A sad loss of many lives occurred in the Iroquois by reason of a blind passageway from the gallery, which led nowhere in particular, but which led out from the main exits in such a way that those rushing outward naturally took it as a line of escape. A few blindly located steps caused some to stumble; others tripped over them, until there was quickly a crowded and confused mass of men, women and children caught in this *cul de sac* at the top of the grand staircase hall and doomed to quick death by suffocation.

Aisles and Exits.

As to the aisles and exits, a great deal of cutting out and enlarging of aisles and removal of seats was done in theaters, in Chicago and all over the country, immediately after the Iroquois fire, apparently without reflection that *to deliver the crowd from the seats at the doorway with too great a rush increases the danger of crushing at the doors and on the stairs.* Indeed, I am of the opinion that the width of the aisles near the stage might reasonably, and with advantage, be made much narrower than the law now permits, thus increasing the number of good seats and the earning capacity of the house enough to pay good interest on the cost of making it safer and providing more numerous aisles, exits and stairways at the rear.

The narrowest aisle permitted in a theatre, even close to the stage, is commonly thirty inches. In a Pullman car and in the ordinary railway coach, twenty-two inches and twenty inches is found ample for a crowd of people moving along with all necessary speed in single file.

It is far better to introduce additional aisles at the expense of making all the aisles narrower, thus lessening the tendency, in a

mad rush, for people to try to crowd past one another, and giving better chance for those who are not strong to steady themselves by holding on with their hands to the seats on both sides the aisle as they go along toward the exit.

I was interested in timing the exit under ordinary conditions, from various representative Chicago theaters after their remodeling and was efficiently aided in this by Mr. Guy C. Shaffer, a junior architect in the office of Pond and Pond. In general we found that from the start of the curtain it was only three and a half to five minutes until the corridors were cleared, with the audience taking all the time needed for leisurely putting on wraps—ordinarily from two to three minutes sufficed for clearing balcony and gallery, and in one minute after the drop of the curtain the aisles of the main floor nearly back to the exits were commonly crowded and continued full until about two minutes after the start of the curtain. The heavy steel curtains took from fifteen seconds to thirty seconds to come down, twenty seconds being the ordinary time.

This time of leisurely emptying must not be taken as being safely sufficient for the same audience to get out if panic-stricken, for crowds may become wedged in to some of the exits and the maxim of making haste slowly may be again forgotten. At the Iroquois, under normal conditions at the close of the performance, there is no reason to think that all in this great crowd could not have found their way safely out in two and a half or three minutes, but starting panic-stricken in the midst of a performance it is different,—the door-keepers may have not opened the gates, or a hurrying crowd may take the wrong path, as to the death-trap in the Iroquois hallway and many other unthought of things are possible, such that, in the design, exits, smoke vents, and automatic sprinklers should each have full, independent, adequate attention and each be independently ready for the worst. At the Iroquois some were still struggling out when the fire chief arrived five minutes after the public alarm, and when he returned at probably nine minutes after the alarm he reports that some were still struggling down from the gallery.

Surroundings or Exposures.

Another feature that is worthy of note before closing is that *it is not essential for safety that a theater should stand in an open lot. Some of the worst theater fires in history have happened*

where the space around the theater was open on three sides or four sides.

It is far more important that attention be given to the detail of fire walls and to providing safe passageways. It should, however, always be the effort that channels of strongly arched masonry, passageways roofed almost as strongly as for a fortification, be provided running in opposite directions, so that if a fire from explosion or other unusual cause be developed in the street or along the main façade of the theater, all of the audience could easily find exit in an opposite direction to the alley or to the adjoining street.

Weekly Inspections.

In safeguarding our factories against fire, we find systematic inspections and the filing of a weekly report one of the very best means toward safety. It would be of equal value for theaters. A printed blank can readily be devised for each particular theater, or one for all the theaters of a given city. This should cover the completeness and operative condition of all valves, fire hose, sprinklers, fire-pails, soda-water extinguishers, pole-hooks, fire doors, exit locks and latches, smoke vents, fire-curtain mechanism, and particularly of the neatness, cleanliness and order of every room, passageway, closet, air chamber, loft, basement and fly gallery, used as a part of the theater building. This inspection should be made on each Monday afternoon, since the week end is the time when attractions are commonly changed and the confusion of new acts and strange properties is most apparent.

A private fire brigade from the regular stage hands and ushers should be drilled regularly, the Monday drill to be a "wet drill," testing the stage hose and a few of the soda-water extinguishers, which may be turned out of the window to the area way, or into some convenient drain provided for the purpose, at the rear of the stage.

The head stage carpenter should always be present during this performance as chief of this theater fire-brigade.

If the municipal ordinance required such reports and drills as just described, and that a duplicate of the report be filed each Monday afternoon with the public fire chief of the district, a single fireman or inspector detailed as instructor to cover in turn all the theaters of a large city would, in my judgment, accomplish more real good than the one or two stage firemen at each theater

CITY OF **THEATRE INSPECTION**

Hours time in Theatre ....
Conferred after insp'n with Mr ....

Name of Theatre ....
.... Inspected by ....
Location .... On .... 1904 ......

SURROUNDINGS—N .... S .... E .... W .... Above .... Below ....

State any specially good or bad features ....

SEATING CAPACITY—Floor .... Boxes .... Balcony .... Gallery .... Total ....

Maximum standing admission allowed—Floor .... Balcony .... Gallery .... Total ....

BUILDING—Gen. type constr. .... Quick burning (Joisted with hollows in floors and walls) .... Semi slow-burning (Joisted or plank, all open or well fire stopped) .... Incombustible.

AUDITORIUM SECTION.

WALLS— Brick. Wood. Hollows behind finish—none. ....
ROOF—Joisted. Plank. Incomb. Hollow. Steep. Flat. ....
FLOORS—**In Bst.**—Wood. Joisted. Incomb. Hollow. ....
In Audt'm—Wood. Joisted. On Incomb. Incomb. Hollow. ....
Floor at lowest seat bank .... ft. above, below street. ....
In Attic—None. Wood. Joisted. Incomb. Hollow. ....
ATTIC— Large. Small. Vacant. Misel's. Storage. ....
AUDITORIUM—Dangerous rooms connecting—None ....
Furns'ngs and Finish—Dangerous. Fair. Good. Excl't. ....
Rise Betw'n Seat Banks—Bal .... In's. Gal .... In's. ....
BSM'T—Occup'd by—Toilet and Smoking Rms. Passages. ....
Plenum chamber. Stage dressing rms. ....

EXITS (Regular and Fire Escapes)—Rear. One side. Both sides. Via stage. ....
...Scant.... Fair.... Ample. Marking—Poor. Fair. Good. ....
Blind passages or doors apt be mist'kn for exits— none. ....
Pass'gs—Obscure. Plain. Dang's steps. Bad, good, stairs. ....
Separation currents people from Floor, Balc'y & Gall'y } Reg. Exits—Poor. Fair. Good. ....
F. Escapes—Poor. Fair. Good. ....
AISLES—More needed. Poor. Fair. Good. ....
EXIT DOORS—Can they obstruct or restrict pas'ge? Yes. No. ....
Fastenings—Dangerous. Fair. Good. Excl't. ....
OUTSIDE FIRE ESCAPES—Iron. Wood. Narrow. Fair. Wide. ....
Subject to fires through windows and doors. Not subject. ....
Strength and Repair—Poor. Fair. Good. Excl't. ....

STAGE SECTION.

PROSCENIUM WALL—Brick. Complete bst. to roof. Inc'mp'e. ....
Size main op'ng .... x .... ft. .... other op'ngs. ....
Fire Doors—At all except main op'ng .... Op'ngs unpr'ctd. ....
Wood, Tinned, Iron—Poor. Fair. Good. Excl't. ....
FIRE CURTAIN—Asbestos. Steel. Bare. Proc'td. ....
Opening between curtain and wall .... inches. ....
Side Guides—Steel cables. Iron grooves .... ins. deep. ....
Suspension—Steel Cables. Pulleys & Battons—iron, wood. ....
Anchorage—Steel cable dead lines. Locking angles at top. ....
Operated from—Stage floor. Fly gal'ry ....

word through which a line is drawn.

WALLS—Brick. Bare ....
ROOF—Joisted. Plank. Incomb. Hollow. Steep. Flat. ....
GRIDIRON—Wood. Iron. Pulleys and Fittings—wood, iron. ....
VENTS—1/ of stage area. Held by, rope, fusible links. ....
Automatic Opening—Quick. Slow. Reliable. Uncertain. ....
GALLERIES—Joisted. Plank. Incomb. Hollow. ....
Stairs—Wood. Iron. Fire escapes—scant, ample. ....
DRESSING RMS.—On Gal'ys. Back of Boxes. Under Stage. ....
Wood. Incomb. Hollow. Fire Escapes—scant, ample. ....
FLOOR— .... Joisted .... Plank .... Incomb. Iron Beams. ....

FIG. 17.

The condition is indicated by [illegible]

Power—Hand. Electric Motor. Hydraulic..................
Dropped—By hand only. Automatic by melting of links.
How Used—Open and close of each perf'mnce. Each act.
As a life-saving device—Poor. Fair. Good. Excl't.

BSM'T—Occp'd by—Dres'g rms. Pass'gs. Nearly vacant.
Storage. Floor—Wood, incomb. hollow..................
EXIT DOORS—Can make dang'rs drafts. Well protected.
SCENERY—Am't. Small. Ordn'y. Large. Not, Fireproofed.

GENERAL.

BOILERS—In Bst., of Adt'm., of Stage. Outside. Not, Cut off.
Power—Heat, Direct, Ind'ct. Steam Pipes—safe, suspicious.
Fans—In Attic. In Bst. Well. poorly arranged..........
PLENUM CHMB'R & Ducts—Poorly, Well cut off fr'm adjc't rms.
Contain dang's comb. Other storage. Vacant, Dirty, Clean.
ELECTRICITY—From Theatre Plant. From Public.
Switch Bd.—Wood, Incomb.—Poor. Fair. Good. Excl't.
Gen'l Wiring—.... ..Open. ....Concl'd..... . Mould'g.
..Conduit. Poor. Fair. Good. Excl't.
Stage Lamps and Cables—Dangerous. Aprv'd Types.
GAS—Used in Audt'm. Dr'ssing Rms. Offices.
On Stage. Flames, poorly, well protected.
EXIT LIGHTS—..................Oil..................Gas.
Electric—Independent. Supplied through, stage sw. bd.
If Ind. how connected? ....
FOYER LIGHTS—Elec. Gas. Ind. Supl'd. thro. stage sw. bd.
If Ind. how connected?..................
WATCHMEN—Service dur'g perfm'es. Poor. Fair. Good. Excl't.
FIRE AXES & Pole Hooks—Supply poor, good ....
THEATRE FIRE BRIGADE—Unorg'zed. Undrl'd. Org'zed. Drl'd.
Probable Efficiency—Poor. Doubtful. Fair. Good. Excl't.
FIRE PAILS—Supply, scant, good. Neglected. Well filled.
PORTABLE Extg's and Pumps—Supply scant, good. Neglected.
Well cared for. Some dry powder extinguishers.

SMALL HOSE—**Audt'm.** Bst. Floor. Bal'cy. Gallery.
Stage—Bst. Floor. Galleries. Gridiron.
Fed from.......... Gal. Tank ....ft. above stage roof.
by........in. pipe from city......lbs. meter......By Pump.
Water Supply—Poor. Fair. Good. Excl't.
AUTOMATIC SPRINKLERS—Kind..................
Stage—Roof, Under gridiron, Galleries, Dress'g rms., Bst.
Audt'm—Attic. Basement..................
Fed from .. ..Gal. Tank ...ft. above highest head.
from........Gal. Press. Tank in ....lbs.
from city main.... ....lbs. Pump..................
Wet. Dry System ....in
Valves to Auto Spr.—All exam'd. All seal'd. All fn'd open.
Arrangem't Spr. Pipes—Poor. Fair. Good.
PUBLIC WATER SUPPLY—Direct pump p. only. Grv'ty dlv'y.
Size st. main.........in. Supply. Poor. Fair. Good. Excl't.
FIRE PUMP—Make ....size......x......x.....
Single, Duplex—Location......
Suction from...... .... Reliable.
Lft. Head ....ft. Supply, scant, ample.
Power during performances—Scant, ample. ......Not, Safe.
Test—........in. str'ms........ft.........inch ....hose.
Revs.........Press.........lbs. G. p. m.........Smooth, Noisy.
PUBLIC FIRE ALARM BOX—Main entrance. On stage. None

GRADING AS A WHOLE

Inspector's judgm't of present conditions compared with best of class.

GENERAL ORDER and NEATNESS—	Dang'rs.	Fair.	Good.	Excl't.
Extent of Auto Sprinklers—	None.	Scant.	Tolerable.	Ample.
Small Hose & Porf. Extg's—	None.	Scant.	Tolerable.	Ample.
Water Supply Available—	None.	Scant.	Tolerable.	Ample.
Aid from Public Fire Dept.—	None.	Poor.	Fair.	Good.

	As to Danger to Life				Quality as Fire Risk			
EXPOSURE—	Bad.	Average.	Slight.	None.	Bad.	Average.	Slight.	None.
CONSTR'C'N—	Poor.	Fair.	Good.	Excl't.	Poor.	Fair.	Good.	Excl't.
OCCUPANCY—	Poor.	Fair.	Good.	Excl't.	Poor.	Fair.	Good.	Excl't.
PROTECTION—	Poor.	Fair.	Good.	Excl't.	Poor.	Fair.	Good.	Excl't.
AS A WHOLE—	Poor.	Fair.	Good.	Excl't.	Poor.	Fair.	Good.	Excl't.

* MEANS SEE RECOMMENDATION ON SHEET ATTACHED

THE OPINIONS EXPRESSED IN THIS REPORT ARE SIMPLY THE BEST JUDGMENTS OF THE INSPECTOR

FIG. 17A.

(The inside of the double sheet is ruled and left blank for descriptive remarks under the following headings.)

RECOMMENDATIONS.

Urgent for Protection of Life—

Urgent for Protection of Building & Contents—

Suggestions for further improvements to make this theatre as safe as reasonable practicable without rebuilding—

REMARKS.

FIG. 17B.

—perhaps ten firemen in a small city or one hundred in a large city—required by law to be present from the public force, doing nothing in particular, at the expense of the theater, and who, from my factory experience, will generally be less efficient than the trained and responsible stage carpenter who is at home.

In other words, let the *law* emphasize *fire prevention* by inspection of neatness, order, and precautions more clearly.

The blank (Figs. 17, 17A and 17B on pages 160, 161 and 162) was developed by Mr. E. V. French (member of this Society and of our Mutual Engineer Corps) and myself along the lines of the Mutual Factory Inspection blank. The chief function of such a blank is to focus the attention of the inspector on the several and manifold sources of danger, and its chief virtue is in thus directing the attention of the inspector to safeguards needed and to a test of the condition of all apparatus, rather than its more apparent purpose of presenting a record of faults. The record is condensed to briefest possible compass that the statements may be more conspicuous, and we have found in

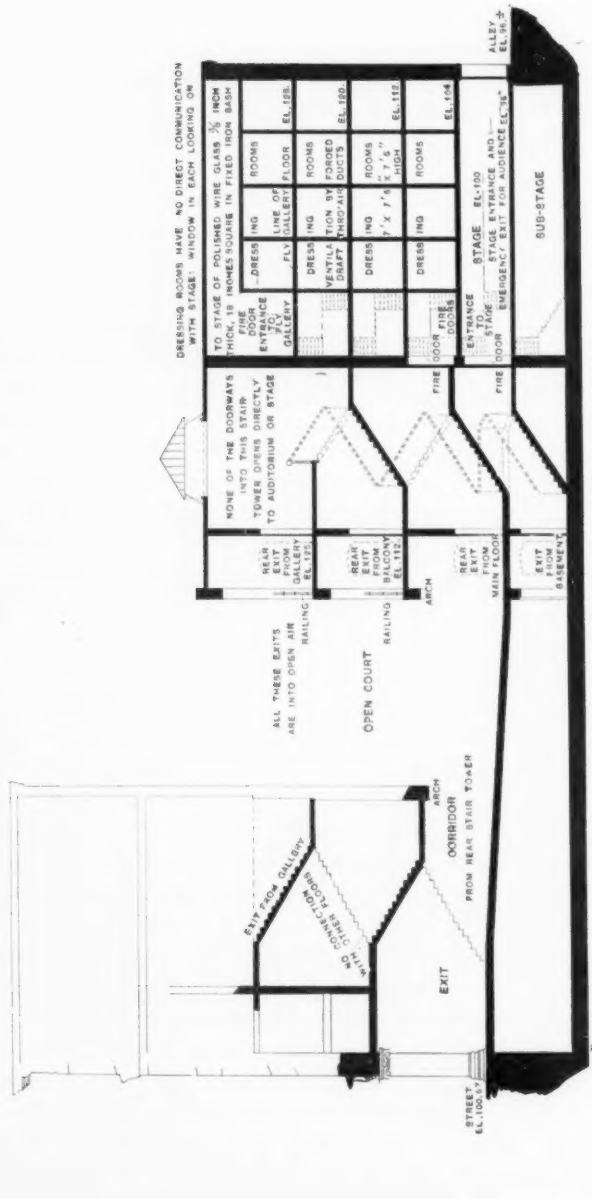
years of factory inspection that brevity in the foundation blank increases the promptness of the remedy. Seventeen Chicago theaters were inspected with this blank in hand, and it seemed to fit fairly well, although it is certain that experience can improve it. I present it here as a convenient summing up of the many points that must be continually looked out for. The condition is shown by an underscore of the word describing the condition found.

For the purpose of illustrating some of the suggestions set forth above regarding the arrangement of exit and stairways, I present on a greatly reduced scale on the pages following some carefully studied drawings that I prepared about two years ago as a means of bringing some of these matters more clearly before certain experienced theater managers, with whom I was discussing certain possible improvements. In the preparation of these plans I also had it in mind to enter a protest against some of the requirements which have been urged by eminent authorities as essential to the safety of the audience, such, for example, as that frequently urged in Europe, that a large theater or house of public entertainment ought to stand in an open lot, and as a means of showing that such arrangements for safety as proposed by the late Sir Henry Irving in his designs for a modern theater were unnecessary.

I therefore purposely assumed the difficulties of a site in the middle of a block, closely built up against on either side and open only front and rear and to the sky above. To make the illustration more complete, I also assumed a minimum width of site. The purpose is to show that the fundamental requirements for safety of the audience and safety of the fire underwriter's risk can all be adequately met on almost any kind of site, and that it is not difficult to provide far more safe and generous exit than is often found.

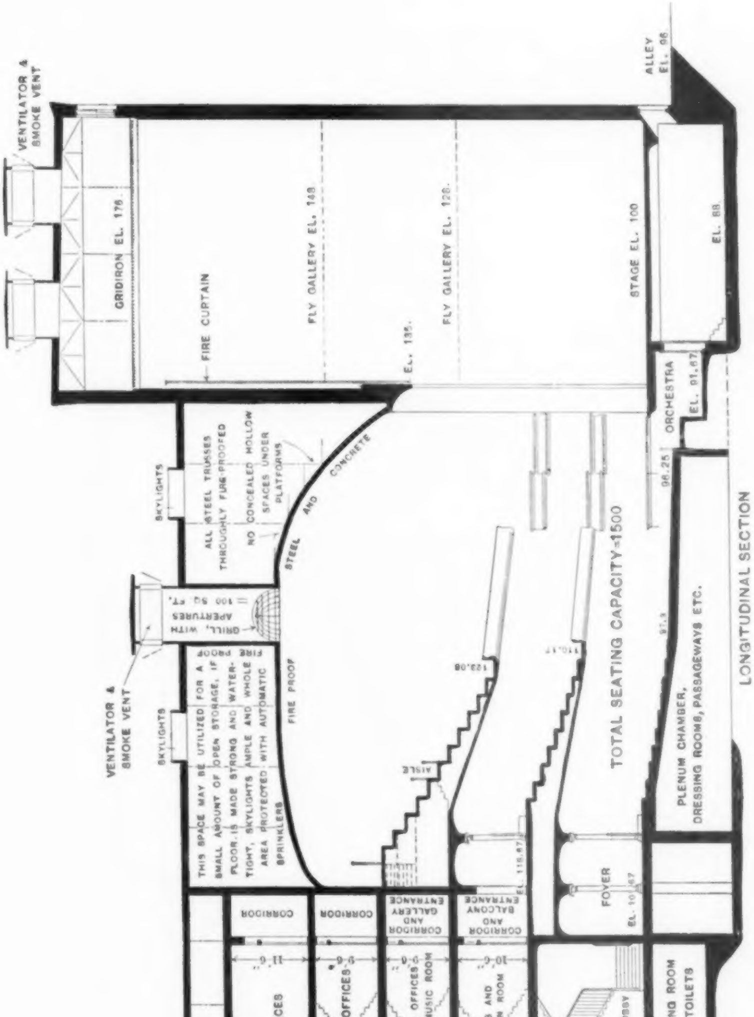
The drawings will set forth the proposed means of providing several exits so clearly that little description is necessary. The total seating capacity is about 1500, a large house. The points of chief interest are:

1st.—The ample exit in four different directions from the balcony and the gallery. I would call particular attention to the exits at the front corners, which have a special value in being always in sight and in front of the sitter, will tend to relieve the crush toward the rear. It was through a small inconspicuous balcony exit thus located that the family of one of my friends



SECTION THRO' STAIR TOWERS (LEFT HAND SIDE).

FIG. 18.



LONGITUDINAL SECTION

FIG. 19.

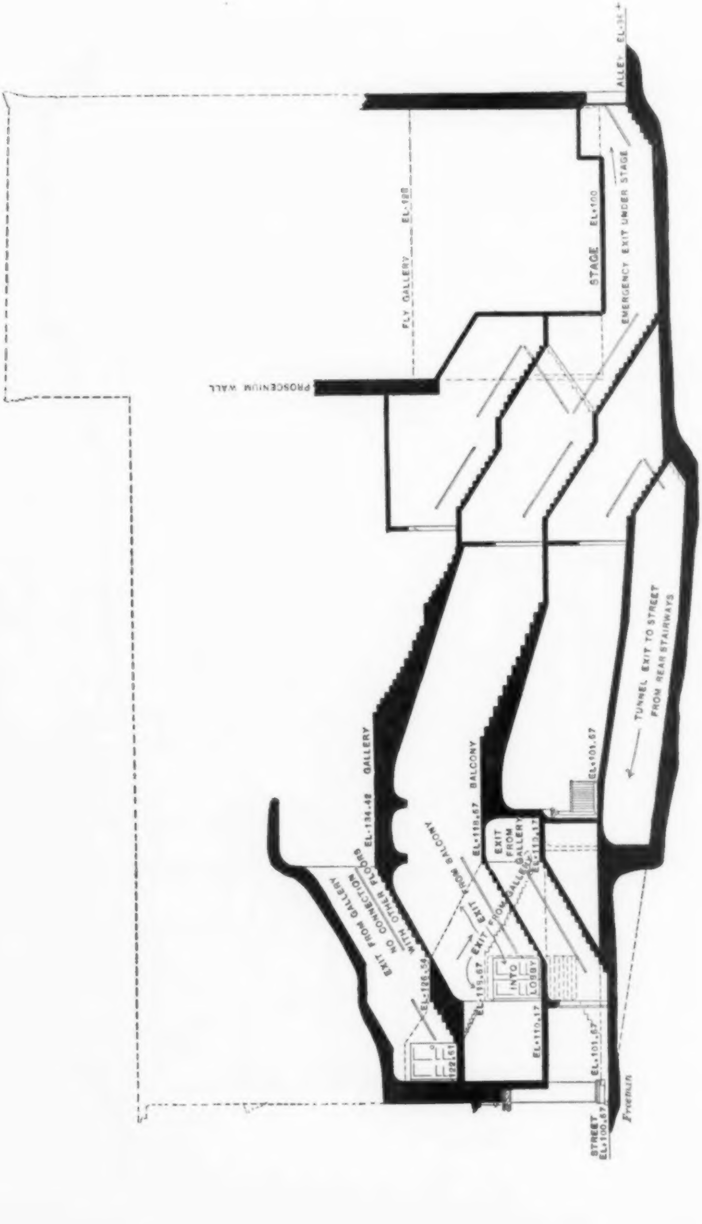


FIG. 20.—SECTION THROUGH STAIRWAY.

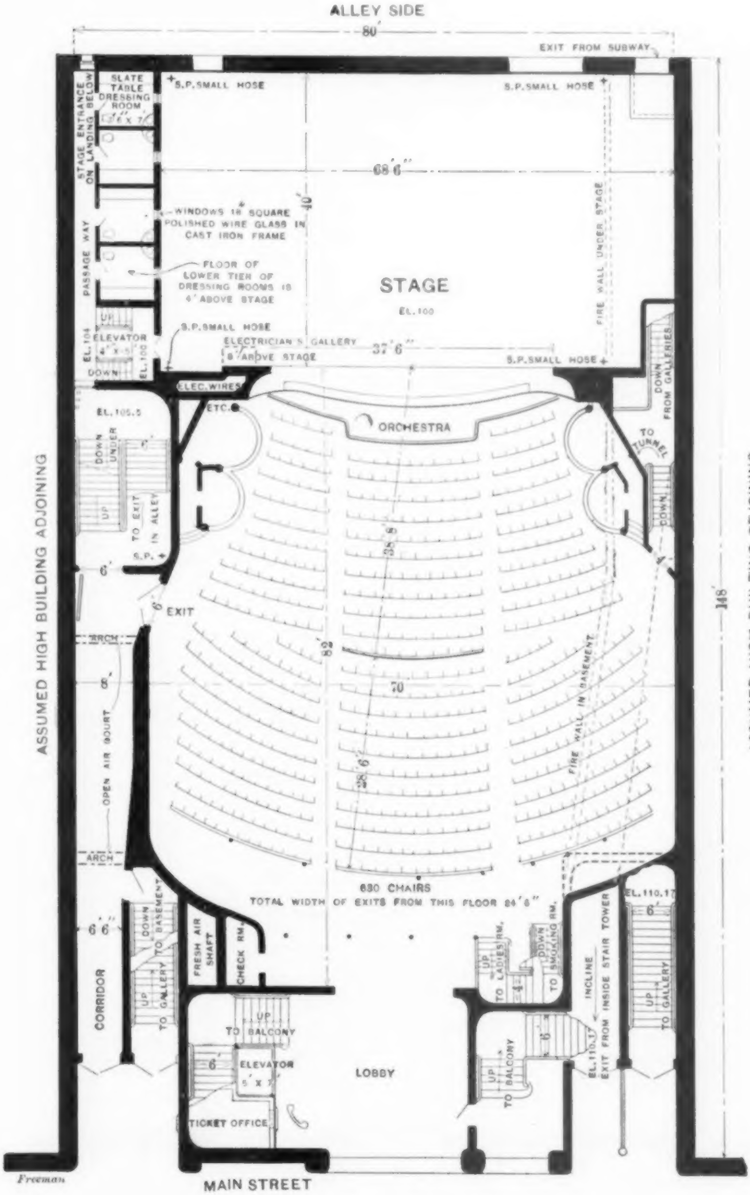


FIG. 21.—MAIN FLOOR PLAN.

AN ILLUSTRATION OF AMPLE SAFE EXITS IN DIFFICULT SURROUNDINGS.

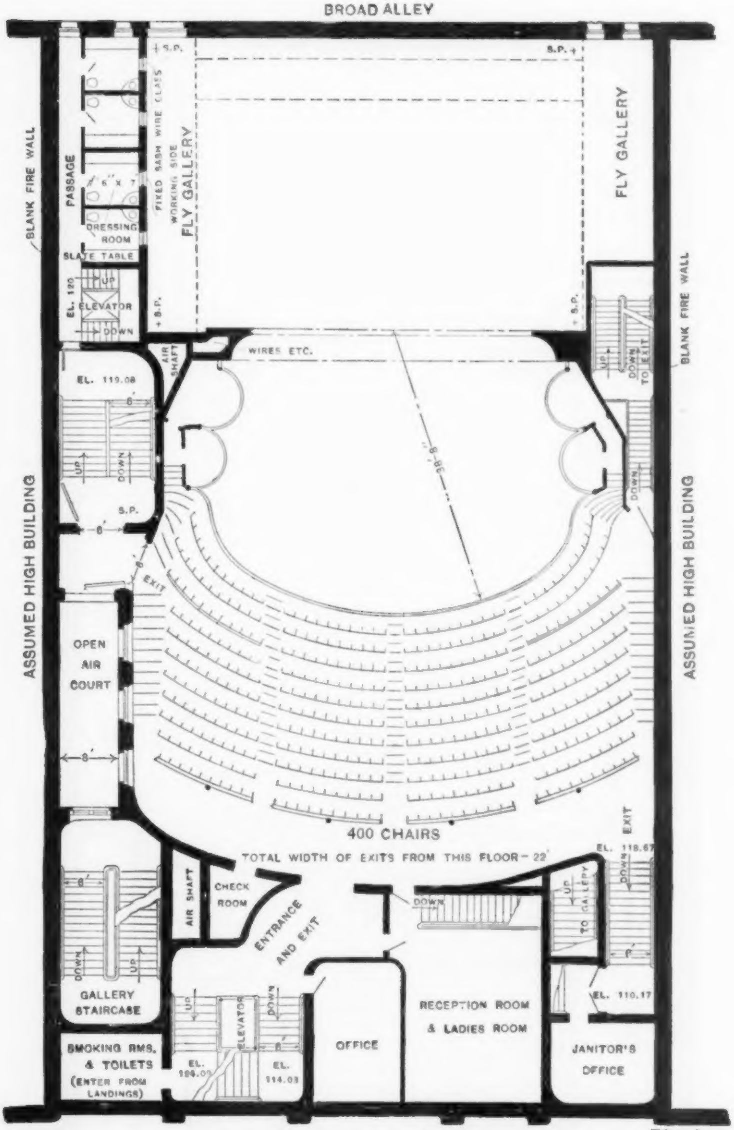


Fig. 22.—An Illustration of Ample Safe Exits in Difficult Surroundings.

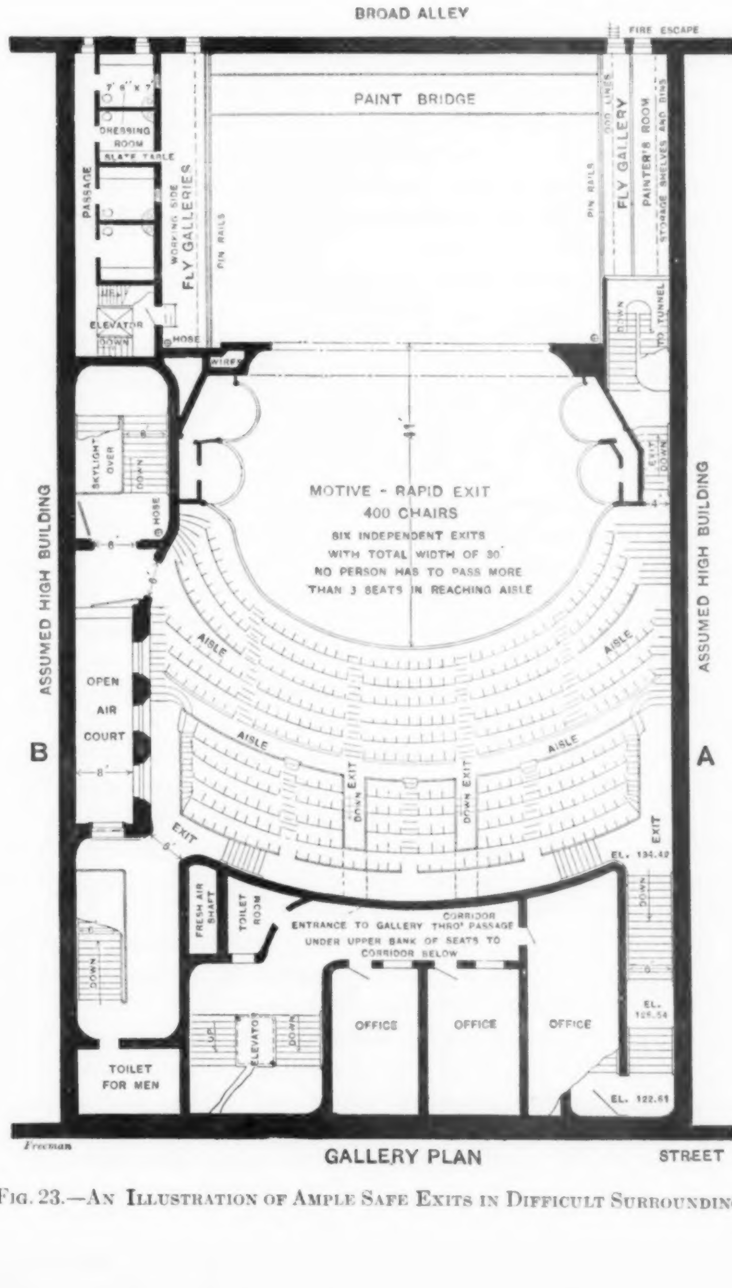


FIG. 23.—AN ILLUSTRATION OF AMPLE SAFE EXITS IN DIFFICULT SURROUNDINGS.

found their way to safety, while the crowd struggled at the rear.

2d.—The use of a tower fire escape (in the rear at the left) modeled on the line of the Philadelphia factory fire escape, communicating with the open air and with no door from auditorium or stage or dressing-room opening directly into the stair-tower proper; it being required that passage be made from the auditorium out across a platform, freely opened to the air, before the stairway can be entered.

This arrangement making it almost certain that the stairway will always be free from smoke.

3d.—Note that the stairway exits from gallery nearest the street are entirely separate from exits from other floors and serve only the gallery. To still further favor rapid exit from the gallery, two additional exits from the middle portion of the seating space drop to a corridor below, making six exits in all, and so scattered that choking about their entrances would appear impossible. As a means of separating the gallery exit from that of the balcony, I have in the spiral layout of the stairs employed a novel device analogous to a double-threaded screw.

4th.—It will also be noted that in view of the enclosed situation two ample exits of large size have been provided to the alley in the rear, for both audience and stage people, each being a sort of fireproof tunnel.

5th.—It will also be noted that provision has been made for permitting daylight to enter the auditorium and stage space, but that the windows can be closed and daylight excluded while an afternoon performance is in progress. These windows should be glazed with prism glass for better diffusion of light if the open-air court is narrow.

By making use of wire glass set in metal frames, and reinforced further by inside shutters folding back into the window jamb, and with automatic sprinklers fed by a large elevated tank, I have no doubt that a building of this type could stand safe in the path of a raging conflagration and thus meet both the best wishes of the fire underwriter and of the humanitarian. This conclusion is given in the light of what I saw in my repeated studies of the ruins after the great Baltimore fire and in the safety of the factory of the Western Electric Co.'s factory in the midst of the great San Francisco fire.

Buildings with their contents *can be made fireproof* by means of automatic sprinklers and adequate protection of the window openings.

No. 1097.*

NATURAL GAS UNDER STEAM BOILERS.

BY JAY M. WHITHAM, PHILADELPHIA, PA.
(Member of the Society.)

1. The writer was recently called upon to give an opinion regarding the commercial operation of a particular mill, both with natural gas and with coal, at certain prices. No detailed data of any special or determinative value could be found in available handbooks and other publications. This paper has been prepared to put on record the results of the author's investigations so as to aid engineers in reaching a fair conclusion on the problem from a commercial standpoint.

Value of Natural Gas Products.

2. The August 7, 1905, Press Bulletin, No. 192, of the U. S. Geological Survey, shows that the value of the present natural gas production of 19 States and Territories of this Union, exclusive of the natural gas produced in Canada and consumed in the States, was $38,496,760 in 1904.

3. In making this figure, Pennsylvania stands first with a product valued at $18,139,914; West Virginia is second, with a product valued at $8,114,249; Ohio is third at $5,315,564; Indiana fourth at $4,342,409, while various other States and Territories make up the balance.

4. The natural gas production, per the Reports of the Director of the Geological Survey, has been valued at

$18,792,725 in 1890
15,500,084 in 1891
14,800,714 in 1892
14,343,250 in 1893

* Presented at the New York Meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

13,954,400 in 1894
13,006,650 in 1895
13,002,512 in 1896
13,826,422 in 1897
15,296,813 in 1898
20,074,873 in 1899
23,698,674 in 1900
27,066,077 in 1901
30,867,863 in 1902
35,815,360 in 1903
38,496,760 in 1904

This shows an increasing production and value, as new fields are opened.*

5. Most of this gas is used for combustion purposes, and much of it for steam production. Hence any information as to the use of such gas in steam generation may be of interest and value.

Composition of Natural Gas.

6. The composition of natural gas varies somewhat with the locality as is shown in the 1903 report of F. H. Oliphant to the Director of the U. S. Geological Survey in a paper entitled "The Production of Natural Gas."

7. The heating values of natural gas per cubic foot, in said report, vary from 1037 to 1287, being referred to 32 degrees Fahr. and 29.92 inches Barometer.

8. Supplemental to the analysis of the West Virginia gas, as given in said report, are the following made for the writer. These tests relate to gas from nine wells in Lewis Co., W. Va., three miles north of Weston, and used on the Cook boiler tests given later on.†

Sample number	1	2	3
Illuminants	0.45	0.15	0.50
Carbonic Oxide	0.00	0.00	0.15
Hydrogen	0.20	0.30	0.25
Marsh Gas	81.05	83.20	83.40
Ethane	17.60	15.55	15.40
Carbonic Acid	0.00	0.20	0.00
Oxygen	0.15	0.10	0.00
Nitrogen	0.55	0.50	0.30
British heat units in a cubic foot of gas at 60 degrees F. and 14.7 lbs. barometer available for useful effect	1030	1020	1026

* Natural gas companies believe these values are very much underestimated.
† See also Vols. IA and 2 of W. Va. Geol. Survey.

9. Also the following analysis of a mixture of natural gas from the fields in the three States supplying Pittsburg, Pa., made in September, 1905, for the writer, may be of some interest:

Illuminants	1.6	per cent.
Carbonic Oxide	1.8	"
Hydrogen	0.3	"
Marsh Gas	81.9	"
Ethane	13.2	"
Carbonic Acid	0.0	"
Oxygen	0.4	"
Nitrogen	0.8	"
	100.0	per cent.

Heat units per cubic feet at 60 degrees Fahr. and 29.92 inches barometer. . . . . 1098 B.t.u.

Testing of Gas Meters.

10. In any test, for any purpose whatsoever, of the use of natural or other gas for a useful effect, it is important that the amount used shall be truly determined. The gas meter can best be tested by the following method employed by the Equitable Meter Company, of Pittsburg, which method appeals to the writer as being complete and exact. It is described by the Company's Engineer, Mr. George W. Barnes, as follows:

11. "The testing apparatus we use, in lieu of a better name, we call a *flow meter,* and it is constructed as follows: First we have an inner cylinder, 24 inches by 24 inches, perforated with as many $\frac{3}{4}$ inch holes as it will contain. Outside of this perforated cylinder we build another cylinder perforated with carefully calibrated holes of $1\frac{1}{2}$ inch diameter, and of a sufficient number to give us a volume sufficient to test a meter using 250,000 cubic feet per hour. The rest of the apparatus is simple, consisting of gauges and a blower of sufficient size to give us the required volume of air at the required pressure.

12. "These holes in the outer cylinder or drum will discharge at a 4 inch water pressure and at a temperature of 60 degrees Fahr. and a barometric pressure of 30 inches, *one cubic foot per second,* or 100 cubic feet in 100 seconds.

13. "Our connections are made up as follows: We connect the meter to the outlet of the blower and our *flow meter* to the outlet of the meter, and operate our blower continuously and govern our

pressure by a regulator placed between the blower and the meter. Our pressure is taken from the outlet of the *flow meter,* and the temperature is obtained at the same point. If the conditions are all as just stated, we maintain the pressure of 4 inches of water pressure. If either the temperature or the barometric pressure vary from the above, we vary our pressure to suit the conditions, thereby maintaining a constant flow of *one cubic foot per second for each hole,* in the shell of the outer drum. We then take time with a stop-watch, allowing the meter to run 100 feet, when we note the time required. If more or less than 100 seconds are required, the error is computed from the 100 seconds, and an adjustment is made in the meter, and the corrections are made accordingly. The run at one 1½ inch hole or outlet will give a test on a meter at a flow of 3,600 cubic feet per hour. After making the above test we remove the stopper of another 1½ inch hole and make the flow at the rate of *two cubic feet per second,* or 7,200 cubic feet per hour, or two revolutions on the 100 foot dial of the meter in 100 seconds, and correct as above noted. This we continue to do in multiples of 3,600 cubic feet per hour until the full rate of capacity of the meter has been reached and adjustments have been made at each period through the run.

14. " This method we think the most nearly perfect to be had, inasmuch as it gives us our tests upon the meter with the actual volumes as required in general service.

15. " The formula used in determining the pressure required with different temperature and barometric pressures was arrived at by making the runs through the orifices from an accurately calibrated *displacement* prover. The experiments covered a period of some eighteen months and have been tried under all of the varying conditions obtainable during that period, while we have also checked back, by the formula, to the barometric reading, temperature and pressure, where each was obtained accurately from instruments, and in all cases we have proved that our formula was absolutely correct for the aforesaid 1½ inch orifices."

Burners to be Used.

16. The writer has investigated several of the gas burners upon the market and concludes that there is but little difference between them from an efficiency standpoint.

17. The following boiler tests made separately with 10 Gwynn

and 10 Kirkwood burners supplying a 250 horse-power Cook water tube boiler, having a "dog-house" furnace, the burners being placed in the front and at a level slightly above the furnace doors, the furnace having an imperfect "check-wall" in each case, and the burners being operated by experts in each case, show that the question of burners may be eliminated.

Test number	790	791
Kind of burners	Gwynn.	Kirkwood.
Duration of tests in hours	8	8
Barometers, pounds	14.25	14.25
Boiler gauge pressure, pounds	126.5	122.03
Draft under damper, inches	0.24	0.1
Gas pressure at burners, ounces	3.84	4.50
Gas temperature at burners, Fahr.	45	45
Chimney temperatures, Fahr.	557	573
Feed-water temperature, Fahr	33.0	33.6
Cubic feet of gas used	95,551	94,388
Cubic feet of gas at 60 Fahr. and 4 ounce pressure	98,348	97,420
Water pumped and evaporated, pounds	70,240	70,030
Boiler horse-power rated	250	250
Boiler horse-power developed	254.5	253.7
Actual cubic feet of gas per boiler horse-power per hour	46.8	46.4
Cubic feet of gas at 4 ounce pressure, 60 Fahr. and 14.25 pounds barometer per boiler horse-power	48.3	48.0

Blue vs. Straw Yellow Flames.

18. The writer has always been led to believe that a blue flame meant the burning of carbonic oxide into carbonic acid gas, although some authors define it as representing the burning of hydrogen into water. Whichever is denoted by the blue flame, no flame of any color is formed at a temperature below about 1000 degrees Fahr.

19. The writer conducted tests on six of the aforesaid 250 horse-power Cook boilers with such a blue flame in the combustion chamber, and, also, complementary tests upon them with a straw-white flame, with the following results:

Kind of flame	White.	Blue.
Test with 4 ounce gas pressure at burners. No.	793	No. 798
No. of 250 horse-power Cook boilers tested	6	6
Steam gauge pressure, pounds	112	88
Draft under boiler damper, inches	0.42	0.52
Pressure at the meter registering the gas used, ounces	18.9	19.8
Gas temperature. Fahr.	46	42

Kind of flame	White.	Blue.
Chimney gases, Fahr.	436	503
Feed-water temperatures, Fahr.	172	178
Cubic feet of gas used at meter pressure	167,420	147,310
Cubic feet of gas used at 4 ounces pressure and 60 Fahr.	182,255	162,210
Water evaporated, pounds	141,566	123,344
Equivalent water evaporated from and at 212 Fahr., lbs.	153,400	132,311
Boiler horse-power made by six 250 horse-power boilers.	1482.1	1278.4
Boiler horse-power made per 250 horse-power boiler	247	213.1
Cubic feet of gas used under actual gas meter pressures and temperatures per boiler horse-power per hour.	37.7	38.4
Cubic feet of gas at 4 ounce pressure and 60 Fahr. per boiler horse-power per hour	41.0	41.0
Openings of gas burner lids	Throttled.	Wide Open.
No. of 6-inch Kirkwood burners used by the 6 boilers .	60	60

These tests showed an advantage in *capacity* in favor of the "White" flame in the furnace at 4 ounces gas pressure at the burners, but no advance in *economy*.

20. Tests were then made at 6-ounce pressure:

Kind of flame	White.	Blue.
Test with 6 *ounce gas pressure at the burners*, test number	794	797
No. of 250 horse-power Cook boilers tested	6	6
Steam gauge pressure, pounds	116	99
Draft under boiler dampers, inches	0.51	0.52
Gas pressure at the meter, ounces	16.5	19.1
Gas temperature, Fahr	44.3	42
Chimney temperature, Fahr.	478	511
Feed-water temperature, Fahr	144	151
Cubic feet of gas used	222,320	185,218
Cubic feet of gas at 4 ounces and 60 degrees Fahr.	241,516	205,424
Water evaporated in the boilers, etc.	165,365	152,600
Equivalent water evaporated from and at 212 Fahr., pounds	184,167	168,333
Boiler horse-power made by six 250 horse-power boilers	1779.4	1626.4
Boiler horse-power made per boiler	296.6	271.1
Cubic feet of gas used under actual gas meter pressures and temperatures per boiler horse-power per hour.	41.6	37.9
Cubic feet of gas at 4 ounce pressure and 60 Fahr. per boiler horse-power per hour	45.2	42.1
Opening of gas burner lids	Throttled.	Open Wide
No. of 6-inch Kirkwood burners used	60.	60

Again, these tests showed a *capacity* advantage in the use of the white flame, but in this case at the sacrifice of *economy*.

21. Tests were then made with the gas at a burner pressure of 8 ounces with the following results:

Kind of flame	White.	Blue.
Tests with 8 ounce burner pressure, test number	795	796
No. of 250 horse-power Cook boilers used	6	6
Steam gauge pressure, lbs	128	128
Draft under boiler damper, inches	0.51	0.54
Gas pressure at the meters, ounces	19.8	18.9
Gas temperature, Fahr.	42	43.3
Chimney temperature, Fahr.	502	508
Feed-water temperature, Fahr	157	184
Cubic feet of gas per meter	183,600	176,510
Cubic feet of gas at 4 ounce pressure and 60 Fahr	203,044	194,004
Water evaporated, pounds	143,864	131,570
Equivalent water from and at 212 Fahr., pounds	158,596	141,319
Boiler horse-power made	1532.2	1365.4
Boiler horse-power per boiler	255.4	227.6
Cubic feet of gas used under actual gas-meter pressures and temperatures per boiler horse-power per hour	40.0	43.1
Cubic feet of gas at 4 ounce pressure and 60 Fahr. per boiler horse-power per hour	44.2	47.2
Opening of gas burner lids	Throttled.	Open Wide
No. of 6-inch Kirkwood burners actually used by the 6 boilers	42	46

These 8 ounce burner pressure tests showed an advantage in both *economy* and in *capacity* in favor of the white flame.

22. Summarizing these six tests in two sets, we find:

Kind of flame	White	Blue.
No. of 250 horse-power Cook boilers used	6	6
Average gas pressure at burners, ounces	6	6
Cubic feet of gas used when reduced to 4 ounces and 60 degrees Fahr.	626,815	561,638
Equivalent water evaporated from and at 212 Fahr., pounds	496,163	441,963
Average boiler horse-power made	1597.7	1423.2
Horse-power made per 250 horse-power boiler	266.3	237.2
Cubic feet of gas reduced to 4 ounce pressure and 60 Fahr. used per boiler horse-power per hour	43.6	43.8
Kind of burners	Kirkwood.	Kirkwood.
Burner lids	Throttled.	Open Wide.

23. The six tests when thus consolidated in two tests show that the *economy* is the same with each and that the *capacity* is greatest with the white flame.

24. Accordingly, the remaining tests by the writer, and named herein, were made with the straw-white flame.

Test of Six 250 Horse-power Cook Boilers.

25. The following tests were made under the most careful conditions upon the same boilers previously referred to and equipped with Kirkwood burners. They were conducted after several weeks had been consumed in "check-wall" duration tests.

Test number	799	812
Duration, hours	16	9
Barometer, pounds	14.3	14.25
Boiler gauge pressure, pounds	120.1	132.7
Draft in front of damper, inches	0.18	0.20
Gas pressure at meter, ounces	16.6	18.0
Gas pressure at burners, ounces	6.9	6.4
Temperature of air	40	69
Fire room	57	73
Natural gas	49	70
Feed water	151.2	185
Chimney	521	494
Gas metered, cubic feet	1,101,350	541,420
Equivalent gas at 60 degrees Fahr. and under 4 ounce pressure with 14.7 pounds barometer	1,179,666	555,617
Water evaporated, pounds	818,700	435,625
Equivalent water from and at 212 degrees Fahr., pounds	906,301	467,948
Boiler horse-power made	1641.8	1507.0
Cubic feet of gas, actual, per boiler horse-power per hour	41.92	39.92
Cubic feet of gas at 4 ounces and 60 degrees Fahr. per boiler horse-power per hour	44.9	40.96
Boiler efficiency, per cent	72.7	.....

Test of a 200 Horse-power Heine Safety Water Tube Boiler at Herron Hill Pumping Station, Pittsburg, Pa.

Heating surface	2,032 sq. ft.		
Shell	42 inches x 21 feet 6½ inches.		
Tubes	116–3½ inches x 18 feet 0 inches.		
Number of natural gas burners in use		6	
Test made	Jan. 7, '05.	Oct. 21, '04.	Sept. 6, '05.
Kind of gas burners used	Kirkwood.	Kline type.	Kirkwood.
Duration of test, hours	10	10	10

Test made by	G. I. Bouton.	Bouton.	Bouton and Whitham.
Test number	....	.....	820
Steam gauge pressure, pounds	145.8	144.1	149.6
Draft in front of damper, inches	0.14	0.18	0.34
Gas pressure at meter, inside of mercury	1.93	2.35	1.81
Barometer inside of mercury	28.54	28.63	28.99
Temperature of external air, Fahr.	25.3	56.1	69
Temperature of fire room, Fahr.	72.8	74.1	78
Temperature of feed water, Fahr.	40.9	61.1	74
Temperature of escaping gases, Fahr	386	450	465
Temperature of natural gas at meter, Fahr.	58.8	70	79.8
Cubic feet of natural gas used	73,720	92,200	105,400
Cubic feet of natural gas reduced to 60 Fahr. and 29.92 inch barometer	....	....	104,521
Cubic feet of natural gas at 32 Fahr. and 29.92-inch barometer	71,199	88,604	98,904
Calorific value of the natural gas per cubic feet in B.t.u. referred to:			
29.92-inch barometer 32 degrees Fahr.	1102	....	....
29.92-inch barometer 60 degrees Fahr.	....	....	1098
Moisture in the steam, part of 1 per cent	0.50	0.53	0.33
Feed water pumped to boiler, pounds	43,800	62,655	74,922
Feed water evaporated, pounds	43,581	62,323	74,672
Equivalent feed water evaporated from and at 212 degrees Fahr., pounds	53,425	75,130	89,024
Evaporation measured in boiler horse-power per hour	154.9	217.8	258.0
Rating of boiler, horse-power	200	200	200
Cubic feet of natural gas used per hour per boiler horse-power developed with gas at:			
29.92-inch barometer and 32 degrees Fahr.	45.97	40.68	38.33
29.92-inch barometer and 60 degrees Fahr.	....	....	40.51
Combined burner and furnace efficiency, per cent	65.8	....	74 92

26. The improvement shown in the last test upon this Heine boiler is largely due to the use of an under and preheated air feed supplemental to the air supply ordinarily carried in through the burners.

Tests of a 302 Horse-power Horizontal Cahall Boiler, at Mansfield, Ohio.

27. Mr. J. Roland Brown contributes the following tests made with six Merrill burners, the boiler having 3,021 square feet of heating surface.

Duration, hours	7	10
Date of test	Mch. 2, '03.	Mch. 5, '03.
Boiler gauge, pressure, pounds	95	85.7
Draft in front of damper, inches	0.28	0.17
Gas pressure, ounces	4.8	7 to 30
Barometer and gas temperatures are not given	....	....
Feed-water temperature, degrees Fahr.	46.5	53.3
Chimney	406	374
Cubic feet of gas used	100,700	88,370
Water evaporated, pounds	67,865	74,168
Equivalent water from and at 212 degrees Fahr., pounds	82,171	89,076
Boiler horse-power made	340.2	260
Cubic feet of metered natural gas used per hour per boiler horse-power	42.26	34

28. The last test was made with gas at a high pressure, varying from 7 to 30 ounces, and the average pressure is not stated.

Burner Tests.

29. Mr. Daniel Ashworth, Member, has contributed the following results of tests made by him with various gas burners on a 2-flue horizontal boiler:

No.	Gas Pressure at Burners in Ounces.	Name of Burner.	Cu. ft. of Natural Gas per hr. per Boiler H.P. Made.
1	0.76	Hoffmann	58.0
2	0.76	Hoffmann	59.7
3	0.34	Reno	67.0
4	0.34	James	63.0
5	2.00	Miller	74.0
6	1.10	Bailey	47.0

Possible Efficiency with Natural Gas.

30. The foregoing constitute all of the natural gas tests that the writer has made, as well as all available and reasonable tests made by others, although there are many tests published by burner people which are wholly impossible.

31. The tests here given show that it is not possible to get better efficiency with natural gas than with coal, and prove rather that the best coal efficiencies can not be obtained with gas. That is due to the large volume of "air for dilution" which must be supplied with gas burners. On the Cook boiler test, No. 799, the Orsat showed as the average of several determinations:

7.8 per cent of CO_2.
8.05 per cent of Oxygen.
0.00 per cent. of CO.
84.15 per cent of Nitrogen.
100.00 per cent. Total.

for the composition of the products of combustion at the boiler damper when 72.7 per cent. of the heating value of the gas was absorbed by the boilers for evaporation.

Such a poor analysis is an impossibility with any boiler when burning coal properly.

32. To produce a boiler horse-power

$$966 \times 34.5 = 33{,}327 \text{ B. t. u.}$$

must be absorbed by the water in the boiler. Assume that, on an average, natural gas has 1,100 B. t. u. per cubic foot. Then, if the gas is burned at 100 per cent. efficiency,

33,327 divided by 1,000 = 30.3 cubic feet

of gas must be used per horse-power per hour. Yet tests are sometimes reported where from only 17.4 to 19.8 cubic feet were used per hour per boiler horse-power.

Assuming 75 per cent. to be the best efficiency obtainable with natural gas under boilers (and this is very difficult), then

30.3 divided by 0.75 = 40.4 cubic feet

of gas, at normal barometer and temperatures, which must be used per hour per boiler horse-power.

Finally.

(1) There is but little advantage possessed by one burner over another.

(2) As good economy is made with a blue as with a white or straw flame, and no better.

(3) Greater capacity may be made with a straw-white than with a blue flame.

(4) An efficiency as high as from 72 to 75 per cent. in the use of gas is seldom obtained under the most expert conditions.

(5) The "air for dilution" is greater with gas than with coal, so that possible coal efficiencies are impossible with gas.

(6) Don't expect, in *good commercial* practice to get a boiler horse-power on less than from 43 to 45 cubic feet of natural gas, the same being referred to 60 degrees Fahr. and 4 ounces pressure above a barometer of 29.92 inches.

(7) *Fuel* costs are the same under best conditions with natural gas at 10 cents per 1,000 cubic feet and semi-bituminous coal at $2.87 per 2,240 pounds.

This is based on 3.5 pounds of wet coal being used per boiler horse-power per hour or 45 cubic feet of natural gas.

(8) Expressed otherwise, a long ton of semi-bituminous coal is the equivalent of 28,700 cubic feet of natural gas; while a short ton of such coal is the commercial equivalent of 25,625 cubic feet.

(9) As compared with hand firing with coal in a plant of 1500 boiler horse-power output, coal being $2.00 per 2240 pounds,—considering labor saving by the use of gas—natural gas should sell for about 10 cents per 1000 cubic feet.

DISCUSSION.

Mr. E. G. Bailey.—In connection with the tests of natural gas under steam boilers, I wish to present two that I made over two years ago, at Fairmont, West Virginia, for the purpose of determining the relative values of gas and coal.

The tests were made on one boiler out of eight similar horizontal return tubular boilers, having 1,002 square feet of heating surface each. These boilers were fitted with grates for burning coal, except at the sides, where they had been removed—making place for Klein burners.

The boilers were tested as ordinarily run, the gas burning with a blue flame and an excess of air passing through the burners, as well as through beds of ashes which had been placed on the grates to prevent the passage of air. Partial data are here given.

Test number	1	2
Duration of test, in hours	8	10
Steam pressure by gauge, pounds	90	88
Barometer, inches mercury	28.92	28.95
Draft in smoke connection	0.20	0.19
Gas pressure at meter, ounce	2.21	2.58
Gas temperature at meter, dry bulb	85	82.4
do wet bulb	63	62.1
Relative humidity of gas	27.5	30.0

Temperature of boiler room, Fahr.	98	104
Temperature of flue gases.	442	443
Temperature of water entering boiler	176.5	164.5
Cubic feet of gas per meters	23,150	32,645
Cubic feet of gas at 32 degrees Fahr. and 29.92 inches mercury; meter correction applied	20,601	29,091
Cubic feet of gas at 60 degrees Fahr. and 4 oz. pressure	21,404	30,225
Density of gas	0.635	0.635
Weight of gas burned, pounds	1,056	1,491
Weight of water fed to boiler, pounds	14,163	20,681
Per cent. of moisture in steam	0.31	0.58
Equivalent water evaporated from and at 212 degrees Fahr. into dry steam	15,172	22,348
Horse-power developed	55.0	64.8
Water evaporated from and at 212 degrees Fahr. per cubic foot of gas at 32 degrees Fahr. and 29.92 inches mercury	.736	.768
Water evaporated from and at 212 degrees Fahr. per pound of gas	14.37	14.99
Cubic feet of gas at 60 degrees Fahr. and at 4 oz. pressure per hour per horse-power developed	48.8	46.6
Analysis of gas by weight:		
Carbon	75.64	75.64
Hydrogen	24.36	24.36
B. t. u. per cubic foot of gas at 32 degrees Fahr. and 29.92 in. mercury	1,199	1,199
B. t. u. per pound of gas	23,410	23,410
Analysis of flue gas by volume:		
CO_2	3.35	4.00
O	14.91	14.05
CO	0.00	0.00
N	81.74	81.95
Per cent. of air excess	232	191
Pounds of air required to burn 1 pound of gas	17.02	17.02
Cubic feet of air required per cubic foot of gas	10.81	10.81
Heat Balance:	Per cent.	
Loss due to latent heat	9.0	9.0
Loss due to pruducts of combustion	7.9	7.8
Loss due to air excess	13.9	11.5
Loss due to formation of CO.	0.0	0.0
Heat utilized in evaporation	59.2	61.8
Radiation and undetermined	10.0	9.9

The above heat balance shows the losses due to latent heat and air excess to be excessive compared with coal. The former is inherent in the fuel, while the latter should be controlled better than in the tests here given.

The reason why higher boiler capacity was not obtained was on account of the capacity of the meter available. It was a new one, made by The Equitable Meter Company of Pittsburg, and the correction used was according to their calibration.

An interesting calculation can be made from the flue gas. As analyzed, the nitrogen is higher than its percentage found in the atmosphere, due to the oxygen combining with the hydrogen, and the moisture formed being condensed and not showing up in the analysis with the Orsats apparatus. A recalculation gives the following:

Test number	1	2
Oxygen forming CO_2	3.24	3.87
Oxygen forming H_2O	3.02	3.25
Oxygen uncombined	14.44	13.58
Nitrogen	79.30	79.30
	100.00	100.00

Now calculating the weights of carbon and hydrogen, using this oxygen, gives a ratio of carbon to available hydrogen in the fuel burned, which in these cases are:

Test number	1	2
Carbon	76.25	78.18
Hydrogen	23.75	21.82

These agree very well with those made in the laboratory with the combustion bulb, which were made for the calculation of the heat balance and neglect the small amount of oxygen and nitrogen which may be present.

Professor E. A. Hitchcock.—The results given by Mr. Whitham are of considerable value, and will be much appreciated by engineers in Ohio on account of the large quantities of natural gas now in use in competition with Ohio bituminous coals. However, in this connection, I can give some comparative results between natural gas and Ohio coal, made upon the same B. & W. boiler in a large lighting plant, each test being of four days' duration with considerable variation in load:

Test on Gas.

Average temperature feed water	133 degrees Fahr.
" steam pressure by gauge	111 pounds.
Cubic feet gas by meter per pounds water	1.32
Pounds water per cubic feet gas	.758
Cubic feet metered gas per hour per boiler horse-power	40.2
Efficiency based on 1,100 B. t. u. per cubic foot natural gas	75 per cent.

Test on Coal.

Average temperature of feed water	139 degrees Fahr.
" steam pressure by gauge	112 pounds.
Pounds water per pound coal (actual)	6.5
Calorific value of coal, B. t. u.	11,960
Efficiency, per cent	59

These results would indicate that in some cases with bituminous coal the efficiency will be lower than with natural gas, and judging from results of several years' experience in testing and investigating plants in this section of the country, I believe such would be the invariable rule. Take the steam plants throughout the country using bituminous coal, and the boilers on an average will not, I believe, show more than sixty per cent. efficiency, and it stands to reason that this efficiency should be higher with natural gas.

I am very much surprised that the analysis of flue gas gave 8.05 per cent. of oxygen. This, to my mind, would indicate a very faultily designed burner or excessive air leaks through the boiler walls. It seems to me that almost ideal conditions should be obtained in the burning of gas with very little "excess air."

In regard to the testing of gas meters, the method described is that used by the Equitable Meter Company at their factory, and this is done by means of air. Now the question is—does the meter give accurate results after shipment and in place, and does it give accurate results after some time of continuous service? We do not know any more than we know that any other piece of measuring or weighing apparatus gives accurate results when in place, unless it is tested on the spot and at the time of the trials. After years of experience in the measuring of natural gas flowing through lines, both in the field and in connection with industrial establishments, I have implicit confidence in the Pitot Meter for indicating rate of flow. When properly applied, its accuracy is without question at all times, and on account of its great simplicity, it can very easily and cheaply be applied for standardizing any meter in place.

Mr. J. Rowland Brown.—In going over the paper by Mr. Whitham, I notice that two tests are reported under my name. Several additions should be made to these tests, and a third test should be added to the set.

The cubic feet of gas per hour per boiler horse-power, at 60

degrees Fahrenheit and 14.7 pounds absolute pressure, has been omitted in both tests. In the 7 hour test, it should be 44.5 cubic feet of gas, and in the 10 hour test 37.4 cubic feet. Taking an average of British thermal units per cubic foot of gas at 60 degrees Fahrenheit and 14.7 pounds absolute pressure, given in section 8, we arrive at a boiler and furnace efficiency of 73 per cent. for the 7 hour test and 87 per cent. for the 10 hour test.

These tests were run on a horizontal Cahall boiler having vertical headers. Another test was run on this same boiler with Gwynn burners. Twelve burners were used, and the following results were secured:

Duration, hours	10
Date of test	5-7-03
Boiler gauge, pressure, pounds	87.2
Draft in front of damper, inches	.17
Gas pressure, ounces	16
Gas temperature (degrees)	45.0
Feed water temperature, degrees Fahr.	54.3
Chimney	475
Cubic feet of gas used	214,220
Water evaporated, pounds	154,115
Equivalent water from and at 212 degrees Fahr., pounds	183,828
Boiler horse-power made ... 76 per cent. above rating	532.8
Cubic feet of metered natural gas used per hour per boiler horse-power	40.2
Cubic feet gas per hour per horse-power, 60 degrees Fahr. and 14.7 degrees pressure	44.2
Efficiency, boiler and furnace	73.4

On this last test, compressed air was used for forcing, and the boiler was operated at seventy-six per cent. above its rating. Results of seventy per cent. efficiency should be every day practice for a boiler operated by natural gas with a properly designed furnace. Such efficiencies are rarely reached with coal firing. There is every reason to believe that efficiencies should be higher in a gas furnace than in a coal furnace.

No trouble is found with deposits of soot in operating a gas furnace, a more uniform steam line is maintained, the fireman can give more attention to the furnace and is not handicapped by fatigue. There should be no large excess of air with a gas furnace, while with hand firing and also with most makes of stokers the excess of air is a serious problem.

Figs. 1 and 2 are two charts showing a steam plant operated

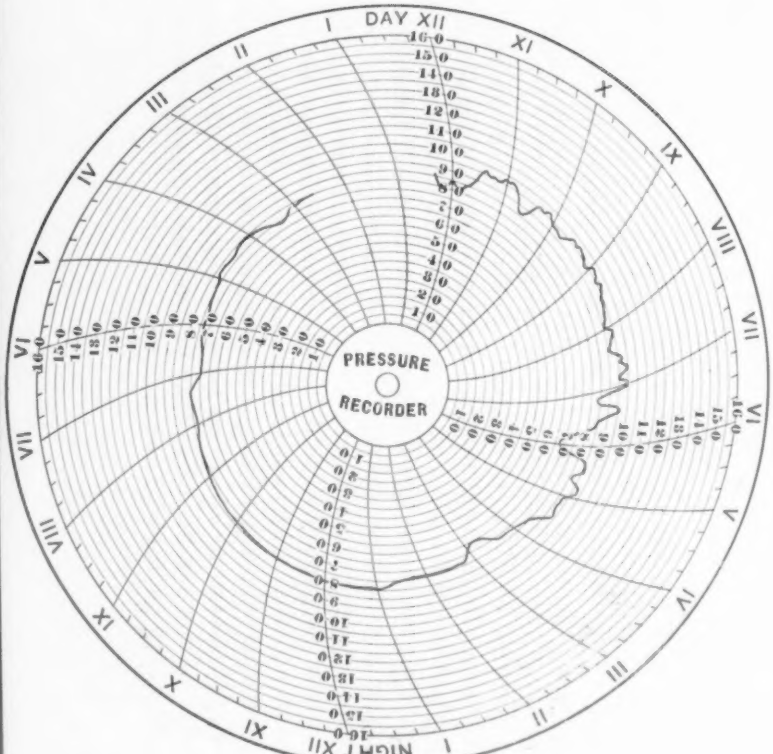


FIG. 1.—WITH GAS.

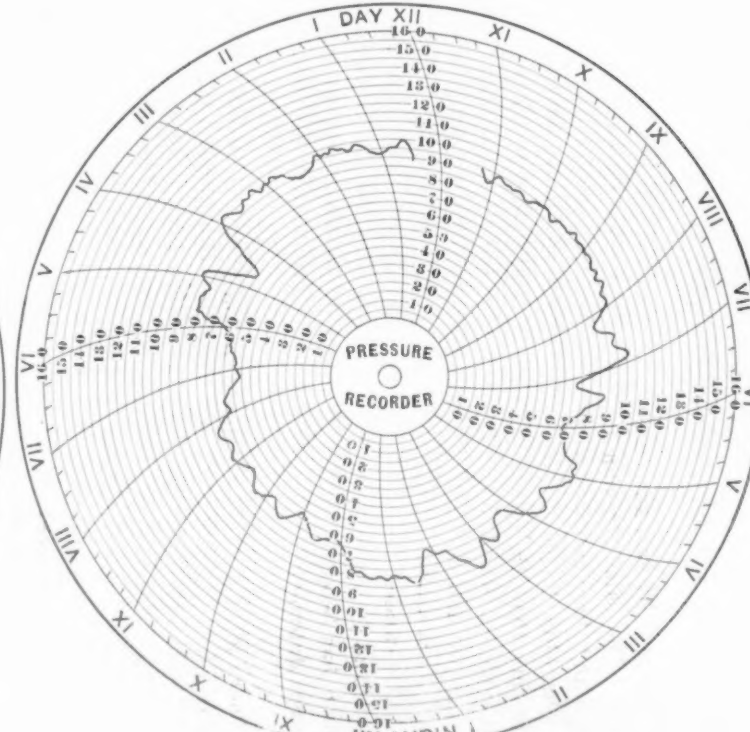


FIG. 2.—WITH COAL.

in one case by gas, and the same plant operated the next day by coal. The same fireman handled the boilers on both of these days and the conditions were similar.

The blue flame is far more efficient both in capacity and economy than the straw colored or white flames. The white flame shows insufficient air. The blue flame is by far the hotter of the two and does not produce smoke, while the straw colored flame does produce smoke. Many persons mistake the reflection of the incandescent brick for the white flame. The blue flame may exist in a furnace, but the reflection from the brickwork, which has become incandescent, makes the flame appear white. These statements the writer has verified within the last two days.

There is very little difference in the efficiency of the various makes of burners now on the market. The greatest difference in efficiency between two different boilers lies in the location of the checker work of the brick in the furnace. The best results are secured with the checker work about three feet beyond the burner. Such a checker work acts as an igniter for the gases with a checker work three feet behind this to break up the flame which passes through the first checker work.

The overhead arch does not make as much difference in a gas furnace as it does in a coal furnace, the checker work being the important factor.

Mr. S. T. Wellman.—Referring to the last paragraph of the paper, I would state that working tests of several weeks' duration made by the Wellman, Seaver, Morgan Co., showed that natural gas at 10.2 cents per thousand equaled coal at $1.55 per ton, plus the labor of firing and handling of coal and ashes.

Professor W. F. M. Goss.—Some years ago, when the supply of natural gas in the Ohio and Indiana fields had so far declined that gas companies could no longer supply their customers at the low prices which had formerly prevailed, the value of natural gas became a matter of importance. Early in the year 1900 I was commissioned by certain parties interested to ascertain the value of gas in comparison with that of other fuels when used for domestic heating. This commission resulted in an elaborate series of experiments designed to give a measure of the amount of heat radiated from stoves of various types when supplied with different fuels. A few results, which may be accepted as representative, are set forth in Tables 1 and 2.

TABLE 1.

PERFORMANCE OF STOVES USING INDIANA NATURAL GAS.

Description of Stoves.	Rate at which Gas is burned.	Performance. B.T.U. Radiated per 1000 cu. ft. of Gas consumed.
Small gas stove without chimney connection ("Omega")	18	851,400
Small gas stove without chimney connection ("Radiator")	16	871,048
Small gas stove with small chimney connection ("Radiator")	22	738,400
Base burner coal stove fitted for burning gas, with normal chimney connections	24	516,716

B. T. U. contained by one thousand feet of gas = 970,000.

TABLE 2.

PERFORMANCE OF STOVES USING SOLID FUELS.

Description of Stoves.	Kind of Fuel.	Rate of Burning. Pounds per Hour.	Performance.—B.T.U. Radiated.	
			For each 2000 pounds of coal burned.	For each cord of Hickory wood burned.
Wood stove having a sheet iron barrel, mounted between a cast iron top and bottom...	Dry Hickory Wood	17		7,651,900
Soft-coal stove having a cylindrical sheet iron barrel mounted between a cast iron fire pot and a cast iron top..	Brazil (Indiana) Block Coal	10	10,472,000	
Hard coal stove, base burner..	Anthracite nut	4	12,471,600	

The Hickory wood used weighed 4,740 pounds per cord.

These results were obtained by means of an air calorimeter consisting of an air tight, insulated chamber or room six feet square, in which the stove to be tested could be set up and fired. The calorimeter had a pyramidal top, ending in a cylindrical opening of known area for the discharge of heated air, and an open space all around the bottom permitted cool air to flow in, and provided means of access for the attendant. All heat radiated from a stove under test was taken up by the air within the calorimeter, which rising from the source of heat, passed out at the top, where its volume and increase of temperature were determined. From these data, the amount of heat delivered from the calorimeter was calculated, which amount, without correction, was assumed to

be the heat radiated from the stove within. The effect of this assumption is to make the stoves appear less efficient than they really were, since in all cases some heat was undoubtedly transmitted through the walls of the calorimeter. The amount thus unaccounted for is known to have been small, it is thought to have been fairly constant, and hence, the effect of its neglect was substantially the same for all tests. As comparative results only were required, absolute accuracy was not needed. To expedite the work, three such calorimeters were simultaneously employed throughout a period of several weeks.

Gas stoves without pipes to chimney must of course be highly efficient as heaters; when applied with chimney connection, their efficiency is at once affected by the chimney loss. A comparison of the value of Table 1, with the number of thermal units in a thousand feet of Indiana natural gas, which may be taken at 970,000, will show that the gas stove making the poorest showing is nevertheless a very efficient device.

The tests of stoves using solid fuels soon disclosed the fact that such stoves, unlike those employed for gas, gave results which varied considerably with variation in the rate of firing. In general the chimney losses were least and the efficiency of stoves greatest when the lightest fires were maintained. As none of the stoves were large, the rates of burning given in Table 2 may be accepted as those of average conditions.

As to conclusions concerning relative values of the several fuels for the purpose of domestic heating, much depends upon the manner in which the gas is burned. Accepting the performance of the Omega stove (Table 1) as a measure of that which may be expected from gas, and the values of Table 2 as a measure of that which may be expected from the several fuels therein described, values may be stated as follows:

In comparison with anthracite coal, gas is worth 6.8 cents per thousand for each dollar per ton charged for coal.

In comparison with bituminous coal, gas is worth 8.1 cents per thousand for each dollar per ton charged for coal.

In comparison with first class hickory wood, gas is worth 11.1 cents a thousand for each dollar per cord charged for wood.

For example, taking values common in central Indiana, in comparison with anthracite coal at $7.00 a ton, gas is worth 47.6 cents per thousand; in comparison with bituminous coal at

\$3.50, gas is worth 28.4 cents; and in comparison with hickory wood at \$6.00 per cord, gas is worth 66.6 cents.

A comparison of these values with those given by Mr. Whitham discloses the fact that gas is far more valuable for domestic heating than for use under boilers. The explanation for this is to be found in the high efficiency of the small gas stove and in the expensive character of the solid fuels for which it is used as a substitute.

Prof. Wm. Kent.—I wish first to speak of the statement concerning the analysis given in paragraph 31. The author says such a poor analysis is scarcely possible with any boiler when burning coal properly. If he means that it is a poor analysis when burning coal, I would say that it is impossible; an analysis so high in nitrogen cannot be obtained no matter how much air is used. The percentage of nitrogen in this analysis is due to the fact that it is natural gas that is burning and not coal.

If he means that it is a poor analysis because there is 8.05 per cent. oxygen in the gas, I would say that it is not a poor analysis at all for burning coal.

If he means that 7 8-10 per cent. of CO_2 in the gas is impossible, an impossibility in any boiler, I would say that that is not so, because you can get any amount from two per cent. up to 16 per cent. Therefore, I would ask an explanation from the author as to what he means by "poor analysis."

Regarding the white flame and the blue flame, as to which some difference of opinion seems to exists, I would say that a blue flame represents perfect combustion and a white flame represents imperfect or delayed combustion at the point where it is visible; but perfect combustion may exist just beyond the white flame. So Mr. Whitham is right in saying that we can get as good economy out of a white as out of a blue flame, provided enough air is supplied to burn the visible white carbon into invisible carbon dioxide, but if the air supply is deficient and the temperature low, the white flame may give off soot, and the economy may be low.

I think he is also right in saying that it is possible to get greater capacity with a white than with a blue flame in some cases, because a blue flame usually means restricted supply of gas and low capacity.

The paper is an admirable one, and the conclusions are generally correct and important.

Mr. John C. Parker.—Mr. Whitham states that there is little difference in burners. It seems to me that this is a broad state-

ment, and it would indicate to me that there was little difference in the particular burners that were used. The explanation of the poor economy might be due to a lack of division; for instance, the economy might be improved by reducing the size and increasing the number of burners for the same capacity.

With respect to the question of the color of the flame I am rather surprised to see that he did not get better results with the white flame. My experience shows that better results are secured from a white flame than from a blue flame.

No. 1098.*

ON THE MEASUREMENT OF AIR FLOWING INTO THE ATMOSPHERE THROUGH CIRCULAR ORIFICES IN THIN PLATES AND UNDER SMALL DIFFERENCES OF PRESSURE.

BY R. J. DURLEY, MONTREAL, CANADA.
(Member of the Society.)

Introduction.

1. While engaged in experimental work requiring the measurement of quantities of air flowing at comparatively low pressures, the writer was led to an examination of the available formulae for the discharge of air under such conditions. These formulae appeared to be insufficient for the purpose in view, and, after some preliminary trials, the experiments described below were made under the writer's supervision, in the Laboratories of the Department of Mechanical Engineering, McGill University, by Messrs. F. E. Sterns and G. M. Smith, Demonstrators in Mechanical Engineering.

2. In these experiments it was desired, first, to ascertain the laws governing the flow of air through circular orifices in thin plates at low heads; and, secondly, to measure the air discharged in such a manner that the experimental conditions could be easily duplicated at any time so that a similar gauging apparatus might be used in the laboratory.

3. Circular orifices bored in thin plates were selected because of simplicity of formation, and it was found that with proper apparatus, such orifices afforded a convenient means of gauging the amount of air discharged by fans, blowers, air engines, air compressors, and the like.

* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

Existing Formulae.

4. The information generally available on the flow of air through orifices at low pressures may be summarized as follows:—

(a) Weisbach's experiments*—These were made on air issuing into the atmosphere, and those made at low pressures gave the values of the coefficient of discharge for orifices 1 and 2.14 centimeter (0.39 and 0.84 inches) diameter when the ratio $\frac{\text{internal pressure}}{\text{external pressure}}$ had the values 1.05 and 1.09. This would correspond, with the barometer at 30 inches, to heads of 1.5 inches and 2.7 inches of mercury, or 22 inches and 37 inches of water nearly, and the coefficient of discharge (c) had values ranging from 0.555 to 0.589 in the formula

$$v = c\sqrt{2gh}$$

where h is the head in feet of air.

(b) Fliegner's equations†—The empirical equation found by Fliegner as expressing results of experiments on the flow into the atmosphere from a reservoir in which the pressure is less than two atmospheres was:—

$$W = 1.060\, F\sqrt{\frac{p_a(p_1 - p_a)}{T_1}},$$

where W = weight passing in pounds per second.

F = area of orifice in square inches.

p_1 and p_a are absolute pressures in pounds per square inch.

T_1 = absolute temperature in degrees Fahr.

Calculation of Ideal Flow of Air Through an Orifice Under a Small Difference of Pressure.

5. The conditions under which the writer's experiments were carried out rendered it advisable to limit the head to 5 inches of water. It is therefore only necessary here to consider the formulae applying to the flow of air under low pressures—

6. Air at 32 degrees Fahr. and at a pressure of 14.7 pounds per square inch weighs 0.0807 pounds per cubic foot. At 60 degrees Fahr. the weight is therefore 0.0764 pounds nearly.

* See Weisbach, "Der Civilingenieur," vol. 5, 1859, p. 546; also, "Theor. Mech.," 464–6 (5th American Edition), and "Peabody's Thermodynamics," p. 137.

† See "Der Civilingenieur," vol. 20, p. 14.

Under a pressure of five inches of water or 0.1505 pounds per square inch above atmospheric pressure, air at 60 degrees Fahr. would have a weight of $0.0764 \times \frac{14.8505}{14.7} = 0.0772$ pounds per cubic foot, an increase in density of 0.0008 pounds, or say, 1 per cent. We shall see that this increase in density is so small that the air may be considered, without serious error, as remaining of uniform density during the experiments here described.

7. The consideration of the adiabatic flow of a perfect gas through a frictionless orifice leads to the equation*

$$W = A\sqrt{2g\frac{\gamma}{\gamma-1}\cdot\frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}. \quad . \quad (1)$$

In this expression

W = weight of gas discharged per second in pounds.

A = area of cross section of jet in square feet.

P_1 = pressure inside orifice in pounds per square foot.

P_2 = pressure outside orifice.

V_1 = specific volume of gas inside orifice in cubic feet per pound.

γ = ratio $K_p/K_v = \dfrac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$ per pound.

For air, where $\gamma = 1.404$, we have from equation (1) for a circular orifice of diameter d inches, the initial temperature of the air being 60 degrees Fahr. (or 521 degrees abs.),

$$W = 0.0816\, d^2\sqrt{\frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{1.425} - \left(\frac{P_2}{P_1}\right)^{1.712}\right]},$$

$$\text{or } W = 0.000491\, d^2 P_1\sqrt{\left(\frac{P_2}{P_1}\right)^{1.425} - \left(\frac{P_2}{P_1}\right)^{1.712}}. \quad . \quad . \quad (2)$$

while, if we measure P_1 in lbs. per square inch,

$$W = 0.07058\, d^2 P_1\sqrt{\left(\frac{P_2}{P_1}\right)^{1.425} - \left(\frac{P_2}{P_1}\right)^{1.712}}. \quad . \quad . \quad (2a)$$

8. In practice the flow is not frictionless, nor is it perfectly adiabatic, and the amount of heat entering or leaving the gas is not known. Hence the weight actually discharged is to be found from such a formula as (2) by introducing a coefficient

* See note at end of paper, also "Peabody's Thermodynamics," p. 132.

of discharge (generally less than unity) depending on the conditions of the experiment and on the construction of the particular form of orifice employed.

9. The above formula is not very convenient for calculation, and for practical purposes it is better (if the ratio $\frac{P_2}{P_1}$ is nearly unity) to employ a somewhat similar expression, obtained by assuming that the air or gas flows as would a liquid of the same mean density. In other words, we neglect the changes of density and temperature occurring as the air passes through the orifice, and are thus able to obtain a simpler though approximate formula—as follows:—

Let i = difference of pressure (between P_1 and P_2) measured in inches of water.

T = absolute temperature of air (supposed to remain unchanged).

u = velocity of air in feet per second (supposed uniform over whole cross section of orifice).

V = specific volume of air (supposed constant during passage through orifice).

P = mean pressure of air in pounds per square foot.

10. Then the head (measured in feet of air) under which the flow takes place is $h = V\ (P_1 - P_2)$.

But a pressure of one inch of water = 5.2 pounds per square foot, so that

$$h = 5.2\ i\ V.$$

Now $u = \sqrt{2gh}$ and the weight discharged per square foot of orifice per second will be

$$\frac{u}{V} = \frac{\sqrt{2g \times 5.2\ i\ V}}{V} = \sqrt{\frac{64.4 \times 5.2\ i}{V}}.$$

But $V = \frac{53.18\ T}{P}$ for air, so that the ideal discharge per square foot of orifice per second will be

$$\sqrt{\frac{64.4 \times 5.2\, iP}{53.18\ T}} = 2.509\sqrt{\frac{iP}{T}} \text{ pounds.*}$$

* Thus when P = 2,117 lbs. per square foot and T is 521° (60° F.) the ideal discharge per second per square foot of orifice will be as follows:

i	1	2	3	4	5	inches of water.
W	5.060	7.155	8.764	10.12	11.312	lbs. per second.

For an orifice of diameter d inches the ideal discharge will be

$$W = \frac{0.7854\, d^2}{144} \times 2.509\sqrt{\frac{iP}{T}}$$

$$= 0.01369\, d^2\sqrt{\frac{iP}{T}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In the usual case, in which the discharge takes place into the atmosphere, P is approximately 2,117 pounds per square foot and

$$W = 0.6299\, d^2\sqrt{\frac{i}{T}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This equation is, of course, of the same form as Fliegner's well known formula.

11. It will be found that up to a pressure of about 20 inches of water (or 0.722 pounds per square inch) above the atmospheric pressure, the results of formulae (2) and (4) agree very closely. At higher differences of pressure divergence becomes noticeable. For example, if $P_1 - P_2 = 60$ inches of water or 2.166 pounds per square inch, we have from (4) (if $P_2 = 2.117$ pounds per square foot and $T = 60$ degrees Fahr.)—for a 3 inch orifice.

$$W = 0.6299\, d^2\sqrt{\frac{i}{T}} = 0.6299 \times 9 \times \sqrt{\frac{60}{521}} = 1.892 \text{ lbs. per}$$

second. In the case of adiabatic expansion, from (2) we get

$$W = 0.0004901\, d^2P_1\sqrt{\left(\frac{P_2}{P_1}\right)^{1.425} - \left(\frac{P_2}{P_1}\right)^{1.712}}$$

$$= 0.00049011 \times 9 \times 2{,}422\sqrt{.8735^{1.425} - .8735^{1.712}}$$

$$= 0.00049011 \times 9 \times 2{,}422 \times 0.1766$$

$$= 1.885 \text{ lbs. per second.}$$

The discharge as calculated by Fliegner's formula would be

$$1.06\, F\sqrt{\frac{P_2\,(P_1 - P_2)}{T}} = 1.06 \times 7.07 \times \sqrt{\frac{14.7 \times 2.166}{521}}$$

$$= 1.06 \times 7.07 \times 0.2471$$

$$= 1.852 \text{ lbs. per second.}$$

Even at the low pressures here considered Fliegner's rule gives results agreeing much more nearly with those of equations (2) and

(4) than with the discharge actually measured. Thus for a 3 inch orifice we have:—

IDEAL DISCHARGE (pounds per second)	Head = 1	2	3	4	5 ins. of water.
By equation (4)	0.248	0.351	0.430	0.496	0.555
By equation (2)	0.248	0.350	0.429	0.495	0.553
Discharge by Fliegner's rule	0.232	0.338	0.415	0.485	0.535
Actual weight of air per second as found from experiments and curves	0.149	0.210	0.257	0.296	0.331

12. It is thus evident that the above formulae for the ideal amount of air discharged require the use of a coefficient of discharge in order that the actual discharge may be computed by their aid, and to determine such a coefficient of discharge for a given orifice, it is necessary to measure the actual discharge and compare it with the amount calculated by such a formula as (2) or (4). These two formulae give results which differ so little within the limits of pressure here dealt with, that the latter has been chosen for use in this paper. Fig. 1 shows the ideal dis-

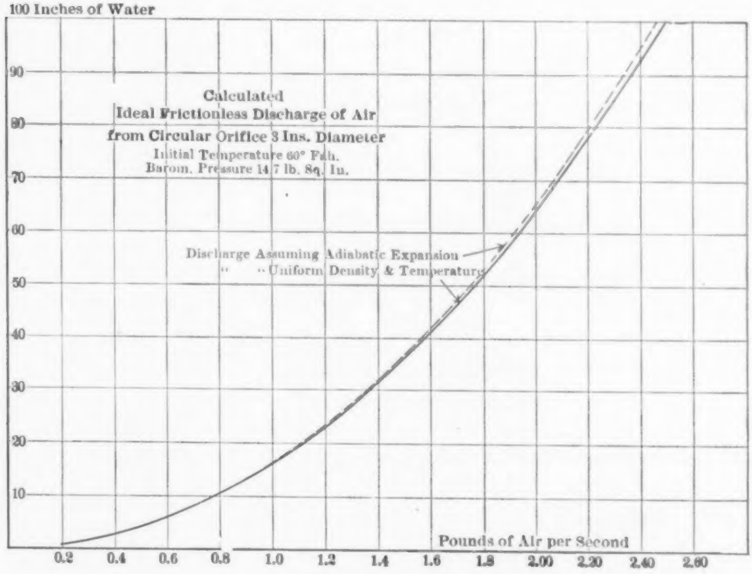


FIG. 1.—COMPARISON OF AMOUNTS OF AIR DISCHARGED ACCORDING TO FORMULAE 2 AND 4.

charge calculated by the two formulae for a 3 inch orifice for various heads; their close agreement, within limits even greater than those of the experiments, is evident.

13. The numbers given in table IV are the values of C in the expression,

$$\text{Actual discharge in pounds per second} = 0.6299\ Cd^2\sqrt{\frac{i}{T}}.$$

It is, of course, understood that they hold good only for orifices of the particular form experimented with, and bored in plates of the same thickness.

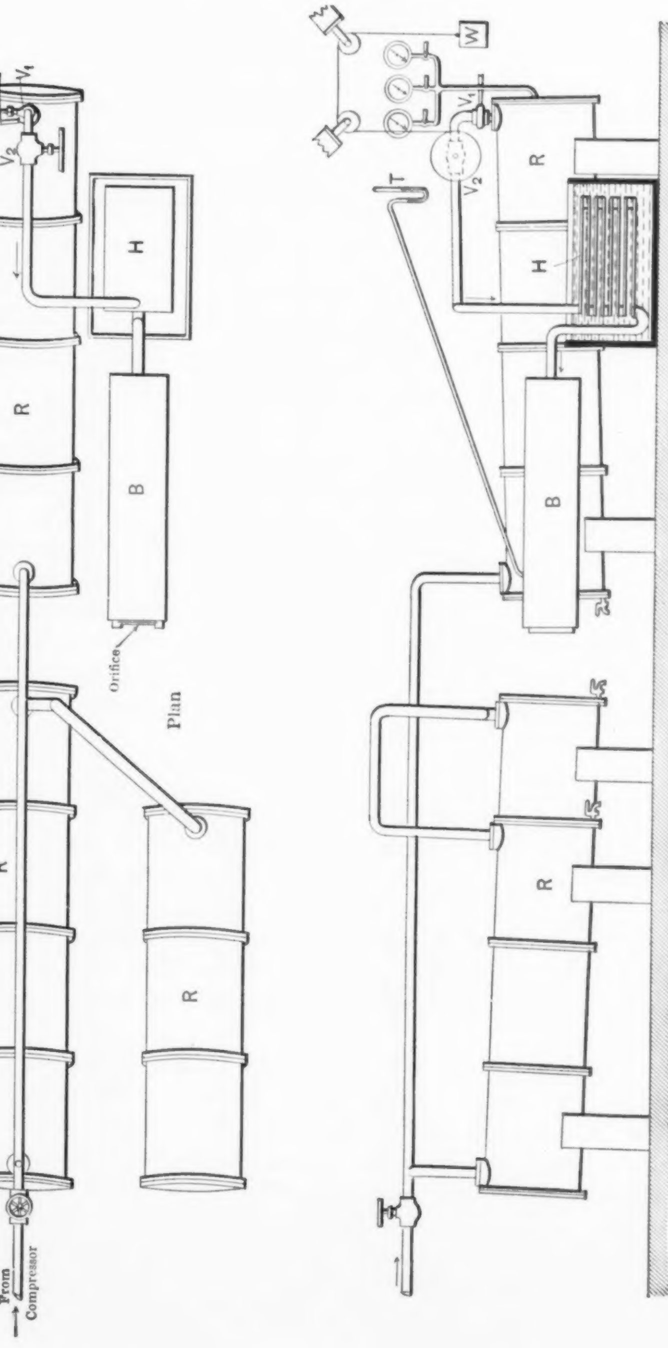
Scope of Experiments.

14. The largest air compressor available was a two stage belt driven machine capable of compressing about 350 cubic feet of free air per minute to a pressure of 100 pounds per square inch.—The air delivered by this compressor was stored in cast iron reservoirs whose combined volume was about 90 cubic feet. With this apparatus it was not found possible to discharge more than $\frac{8}{10}$ of a pound of air per second without getting too rapid a fall of pressure in the reservoirs, and it was accordingly determined to limit the experiments to orifices less than 5 inches diameter and to work only with differences of pressure up to 5 inches of water. The data obtained, however, made it possible to continue the curves with reasonable accuracy so as to give coefficients of discharge for orifices up to, say, six inches diameter and for differences of pressure up to six inches of water.

Description of Apparatus and Methods of Experiment.

15. The general arrangement of the apparatus is shown in Fig. 2. The reservoirs R discharged into the gauging box B through a quick closing gate valve V_1 and a regulating valve V_2. After leaving these valves the air passed at low pressure through a heater H, consisting of a number of thin sheet iron boxes immersed in a tank of water whose temperature could be varied as desired.

16. The gauging box, shown in detail in Fig. 3, was of 1 inch pine made air tight by pitch on the inside. It was 6 feet long and 10 inches by 10 inches inside dimensions. For the experiments on the larger orifices a second gauging box, of similar design, but 16 inches by 16 inches inside, was used. Near the end at which the air entered, suitable baffle plates were placed as shown. The connection for the U tube for measuring the head was placed near the middle of the box.



Elevation

FIG. 2.—GENERAL ARRANGEMENT OF APPARATUS.

17. The air reservoirs (*R*, Fig. 2) were built up of cylindrical flanged castings connected by bolts. The joints were made with asbestos millboard, soaked in boiled oil, and as it was necessary to use fairly high pressures in the reservoirs in order to get

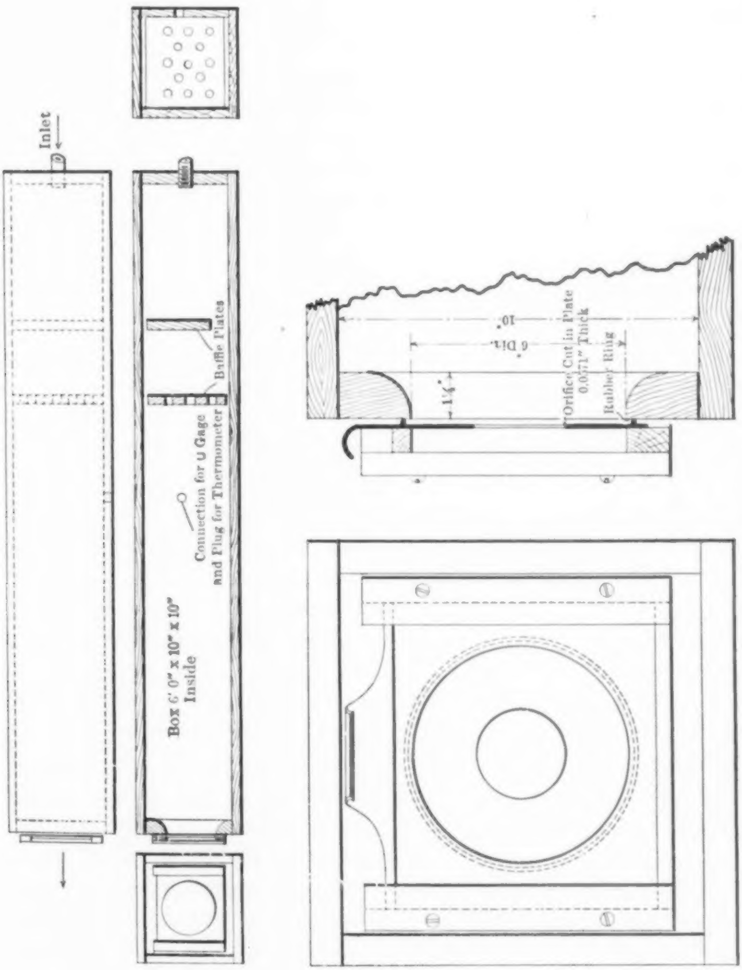


FIG. 3.—DETAILS OF GAUGING BOX AND ORIFICE.

experiments of sufficient length on large orifices, some difficulty was experienced in keeping the joints tight. The leakage was carefully measured and it was found that at the highest pressure it did not exceed 0.0005 pounds per second, while at low pressure

its amount was negligible. The results of the leakage tests were correct to within at least 10 per cent., and therefore the greatest possible error made in allowing for leakage would be 0.00005 pounds per second, not more than 3 per cent. of the whole discharge on the $\frac{5}{16}$ inch orifice, and decreasing to $\frac{1}{100}$ of 1 per cent. on the 4 inch orifice.

18. The orifices were carefully bored in iron plates No. 15 Brown & Sharpe Wire Gauge (0.057 inch thick), and were true to size within $\frac{5}{1000}$ inch. In calculating the ideal discharge, the size measured correctly to within $\frac{1}{1000}$ inch by a vernier caliper, was used in each case. The effect on the result due to inaccuracy in measurement of orifice diameter would therefore not exceed $\frac{7}{10}$ of 1 per cent., on the $\frac{5}{16}$ inch orifice, while on the 1 inch orifice it would be less than $\frac{1}{5}$ of 1 per cent., and on larger orifices would be smaller still.

19. The orifice plates were secured to the end of the box in such a way as to be easily removed and interchanged. The joint between plate and box was made tight with rubber cord.

20. The temperature of the air in the box was measured by a thermometer placed near the middle of its length. This reading was checked at intervals by means of a thermometer suspended lengthwise in the jet, and was never found to vary more than one-tenth of 1 per cent. from the actual absolute temperature in the orifice, except in the high temperature trials. In these trials three thermometers placed at different distances from the orifice were used and from their readings the actual temperature in the orifice was deduced.

21. Standard mercury thermometers were used to measure the temperature of the air in the reservoir. They were inserted through rubber packed stuffing boxes so that the bulb came into actual contact with the air and was therefore exposed to the pressure in the reservoir. This pressure increased the reading about 1 degree for every 100 pounds, which was allowed for correctly to within 0.05 degree. The thermometers were graduated in degrees and tenths Fahr., and were such that temperatures read from them were correct to within 0.1 degree. The error on 500 degrees (the approximate absolute temperature) would therefore be less than 0.03 of 1 per cent.

22. For measuring the pressure in the gauging box, the U tube T (Fig. 2) was placed in a convenient position for the eye of the person operating the regulating valves and was connected

to the box by rubber tubing. A scale of half inches divided on blackened glass was placed behind one leg of the U tube so as to read inches of head, and was illuminated by an incandescent light suspended behind it. To avoid parallax the scale was read through a small telescope fastened with a clip so as to slide up and down the leg of the U tube. The scale was graduated correctly to within $\frac{2}{1000}$ inch; the resulting possible error would therefore not exceed $\frac{4}{10}$ of 1 per cent. on the 1 inch head and would be about $\frac{1}{10}$ of 1 per cent. on the 4 inch head. As the meniscus was magnified considerably by the telescope, the error in reading its position would probably not be more than $\frac{2}{1000}$ inch, leading to an error of the same amount as that just mentioned, in the final result.

23. The pressure in the reservoir was measured by Standard Crosby gauges which were carefully calibrated on the dead weight gauge tester. In trials on the smaller orifices a gauge graduated in pounds per square inch was used. This gauge could be read correctly to within $\frac{1}{10}$ of a pound per square inch, parallax being avoided by aligning the pointer with its shadow thrown on the dial by a small lamp suspended opposite the center of the gauge. Where the difference between the initial and final pressures was not less than fifty pounds per square inch, the error would not exceed $\frac{1}{5}$ of 1 per cent. As it was very difficult to keep the head constant when the reservoir pressure became low in trials on the larger orifices, the range of pressure was shortened for these trials, and gauges graduated in tenths of a pound were then used, giving readings correct to 0.02 pound, so that the error on a range of twenty pounds would not exceed $\frac{1}{10}$ of 1 per cent.

24. The reservoirs were provided with drain cocks so that water condensing from the vapor in the air pumped in could be drawn off. The drains were opened so frequently that never more than a pound of water was collected, which would make an error of less than $\frac{1}{40}$ of 1 per cent. on a reservoir volume of 90 cubic feet.

25. In certain trials on the smaller orifices, a small wrought iron reservoir (volume 4.294 cubic feet) was used with the object of getting a wide range of pressure in a short time, and also of checking results obtained from the large reservoir. From this reservoir there was no leak. The other errors were of the same order as those previously mentioned.

26. As mentioned above, some difficulty was experienced in keep-

ing the pressure uniform in the gauging box when the discharge was great. This was partially remedied by stopping the trial before the pressure in the reservoir had fallen too low. It was almost completely obviated by attaching to the spindle of the regulating valve a grooved pulley and leading a cord from its rim over guide pulleys to a weight (W, Fig. 2) of about ten pounds. It was then found that the valve could be opened much more uniformly by resisting the action of the weight than by moving the valve wheel directly. By this means the fluctuations were reduced to less than 2 per cent. of the head, and the trials were so long that the error in measuring the head was substantially only that already mentioned due to inaccurate reading and to imperfect graduation of the scale.

27. Time was measured on some of the trials by an ordinary watch, on others by a stop watch, and on others by a recording chronograph. The ordinary watch was used on trials of 5 minutes duration and longer, and as it could be read correctly to within half a second, the error in an experiment lasting 20 minutes would not exceed 0.05 of 1 per cent. On trials of two or three minutes the stop watch was used with an error less than $\frac{1}{6}$ of 1 per cent. The chronograph, which made its records automatically, was used on trials of less than two minutes duration and gave readings correct to within $\frac{1}{50}$ second. As the shortest trial occupied at least 30 seconds, the error would not be greater than $\frac{1}{15}$ of 1 per cent. The trials were started and stopped by means of a quick opening gate valve V (Fig. 2), placed between the regulating valve and the reservoir, the chronograph circuit being closed as the handle of the gate valve passed its mid position.

28. The volume of the reservoir was measured by weighing the amount of water necessary to fill it. The water was weighed in tanks holding 100 pounds, on scales reading correctly to half an ounce, so that the error could not exceed $\frac{1}{16}$ of 1 per cent. The total volume of the three parts of the reservoir and the piping was found to be 91.881 cubic feet. The various errors just discussed are enumerated in Table 1.

29. The method used in making the trials was as follows:— The reservoir was pumped up and the valve between it and the compressor was then closed. Sufficient time was allowed for the air in the reservoir to come to a uniform temperature. The gate valve V_1 being open, the regulating valve V_2 was opened till the head of air in the box was brought to the point at which the

trial was to be run. The gate valve was then closed and the chronograph switch thrown in so that when the valve was reopened a record of the time would be made. The initial pressure and temperature in the reservoir were then read and the trial started. The final temperature and pressure were not read immediately at the end of the trial as it was found that some time elapsed before the air in the reservoir had resumed a uniform temperature. The leak during this time was allowed for, but was very small since the pressure in the reservoir was low.

Accuracy of Results.

30. Upon consideration it was found that the effect of the errors shown in Table 1 upon the final result might cause a total possible error of about $\pm$ 7 per cent. on the $\frac{5}{16}$ inch orifice at 1 inch head, or of about $\pm 1\frac{1}{2}$ per cent. in the case of the 4 inch orifice at 4 inch head. The actual deviations of the observed results from the mean curves were found to be within these limits when allowance was made for the effect of the size of the gauging box on the results, as explained below.

31. The probable errors have been calculated for the various series of experiments made at different heads, and are,—

Head.	Probable error %.
1 inch.	0.735
2 "	0.470
3 "	0.635
4 "	0.667
5 "	0.650

It seems permissible therefore to regard the figures of Tables 4 and 5 as being reliable to within less than 1 per cent.

Corrections for Barometric Pressure and Temperature.

32. From equation (3) it is seen that when the difference of pressure on the two sides of an orifice is small, the weight of air discharged per second should vary directly as the square root of the mean pressure and inversely as the square root of the absolute temperature. In tabulating the results of the experiments made it was therefore necessary to reduce all the actual discharges, as measured, to their values at 60 degrees Fahr., and 30 inches barometric pressure. A special series of trials (Nos. 151-162) was made to see whether the actual discharge would vary as above

with regard to temperature. For this purpose the air was passed through the heater (*H*, Fig. 2) previously described, and was delivered to an orifice one inch diameter under a head of 3 inches of water, at a series of successively increasing temperatures. It was found during each experiment that the temperature in the gauging box could be kept constant within 1 degree Fahr., and the trials were made between the limits 55 degrees and 105 degrees Fahr. After measuring the actual discharge in each case, and correcting for barometric pressure, the mean line (Fig. 4) was

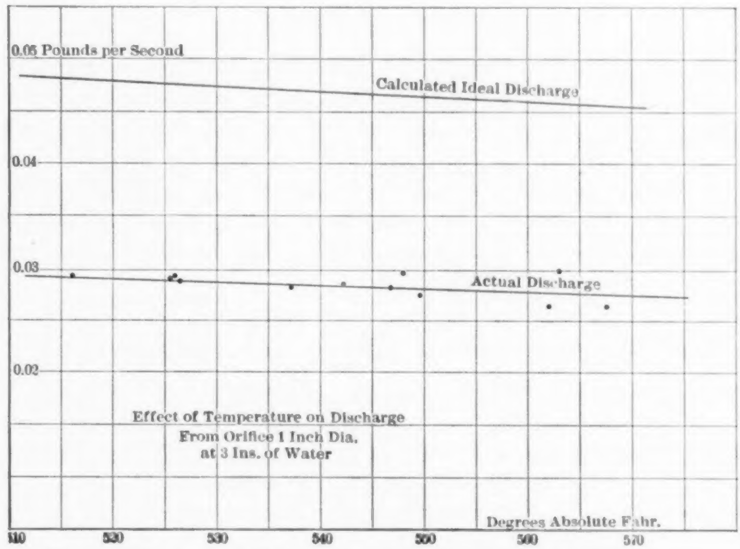


FIG. 4.—EFFECT OF TEMPERATURE ON DISCHARGE FROM ORIFICE.

drawn—and the corrected discharges tabulated from it. The figures for these trials are given in Table 2, and show that within the limits of these experiments the discharge may be taken to vary inversely as the square root of the absolute temperature in the gauging box.

33. It was found impossible to make the numerous joints of the heater perfectly air tight. The air leaking out was, of course, visible in the water in the containing tank, its greatest amount was found to be about 0.000037 pounds per second and was allowed for in calculation, being measured by displacement in each case. The heater was not employed for the other trials and in them this source of error did not exist.

Calculation of Actual Discharge from Experiments.

34. Trials 1-146 were made with the temperature in the box as near 60 degrees Fahr. as possible, and variations from this temperature were allowed for, as mentioned above, on the assumption that the discharge is inversely proportional to the square root of the absolute temperature. Proper correction for barometric pressure was also applied, and from these experiments the actual discharge was then calculated. The corresponding ideal discharge

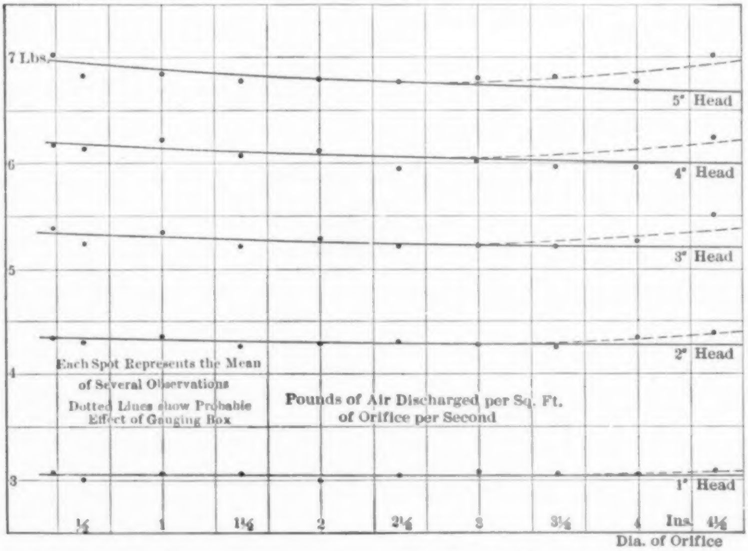


FIG. 5.—AIR DISCHARGED PER SQUARE FOOT OF ORIFICE PER SECOND.

was also worked out from equation (4) for a temperature of 60 degrees and a pressure of 30 inches of mercury. The actual discharge per square foot of orifice was then computed for each experiment and the results plotted on Fig. 5. In Figs. 6 and 6a are shown the values of the coefficient of discharge obtained from the curves of Fig. 5, and from the corresponding ideal discharges.

35. The volume of the reservoirs and attached piping being 91.881 cubic feet, it is readily shown that the weight actually discharged per second (if the pressure in the reservoirs changed from P_1 to P_2 pounds per square inch, and the temperature from

T_1 to T_2 degrees Fahr., absolute, in t seconds) is given by—

$$W=\frac{248.3}{t}\left(\frac{P_1}{T_1}-\frac{P_2}{T_2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The actual discharge was in every case calculated by this formula from the observed quantities, corrections being applied as explained above.

36. With each orifice five sets of experiments were made at heads of from 1 inch to 5 inch of water. Figs. 8-17 (inclusive) show for each orifice the corrected actual discharges and the mean curves plotted on a base of difference of pressure. The mean curves were, of course, found by using the coefficients of discharge of Figs. 6 and 6a.

37. Fig. 6 shows the curves of coefficients of discharge for

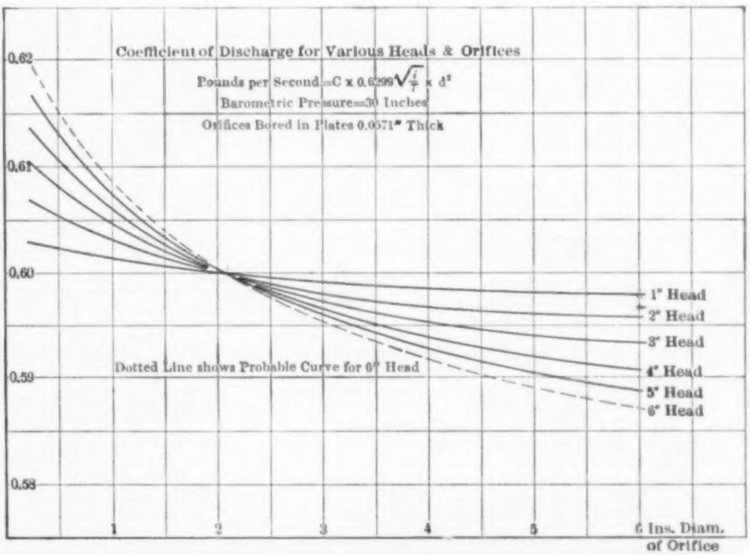


FIG. 6.—COEFFICIENT OF DISCHARGE FOR VARIOUS HEADS AND ORIFICES.

various heads plotted on a base of diameter of orifice. From this diagram and from Fig. 6a can be selected a suitable coefficient for use in any particular case. The detained results of the various trials are given in Tables 6 to 16.

Effect of Size of Gauging Box.

38. From the curves of Fig. (5), it was seen that the actual discharges of the larger orifices at the higher heads were consid-

erably above the mean curve. (See also Figs. 15, 16 and 17.) This seems to indicate that the results were being affected by the velocity of approach of the air. To check this, experiments (147-150 inclusive) were made with a 3 inch orifice at 1 inch and 5 inch heads, using the small box whose area was only 13.8 times that of the orifice. The results of these trials were compared with those obtained from the same orifice in the large box and it was found that the discharge with the small box was 0.2 per cent. higher at the 1 inch head and 10 per cent. higher at the 5 inch

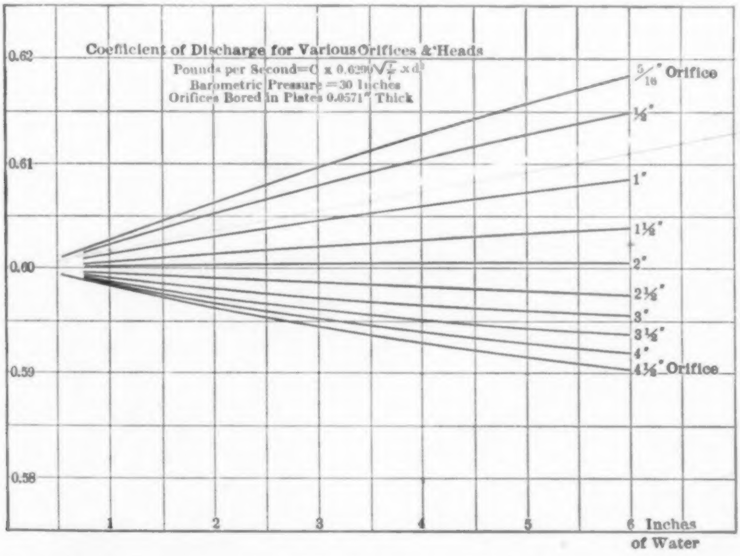


FIG. 6A.—COEFFICIENT OF DISCHARGE FOR VARIOUS ORIFICES AND HEADS.

head. This variation was greater than that observed from the trials of the 4½ inch orifice in the large box, the difference being due to the smaller ratio between the areas of box and orifice. The curve of observed discharge from the 3 inch orifice in the large box, even at the highest heads experimented with, lies only very slightly above the curve of corrected discharge. These results show that the area of cross section of the gauging box should, for heads up to 5 inches, be at least twenty times the area of the orifice if reasonably accurate results are to be obtained. For greater heads a still larger box would probably be necessary.

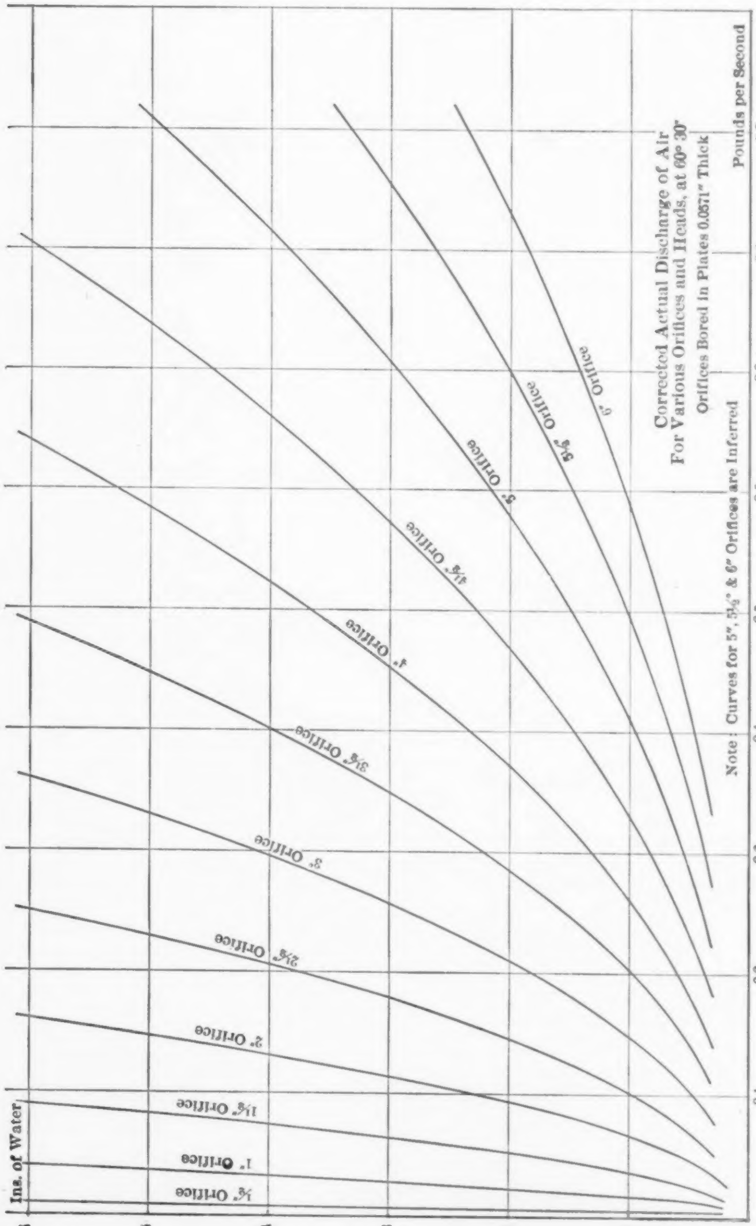


FIG. 7.—CURVES OF CORRECTED ACTUAL DISCHARGE OF AIR.

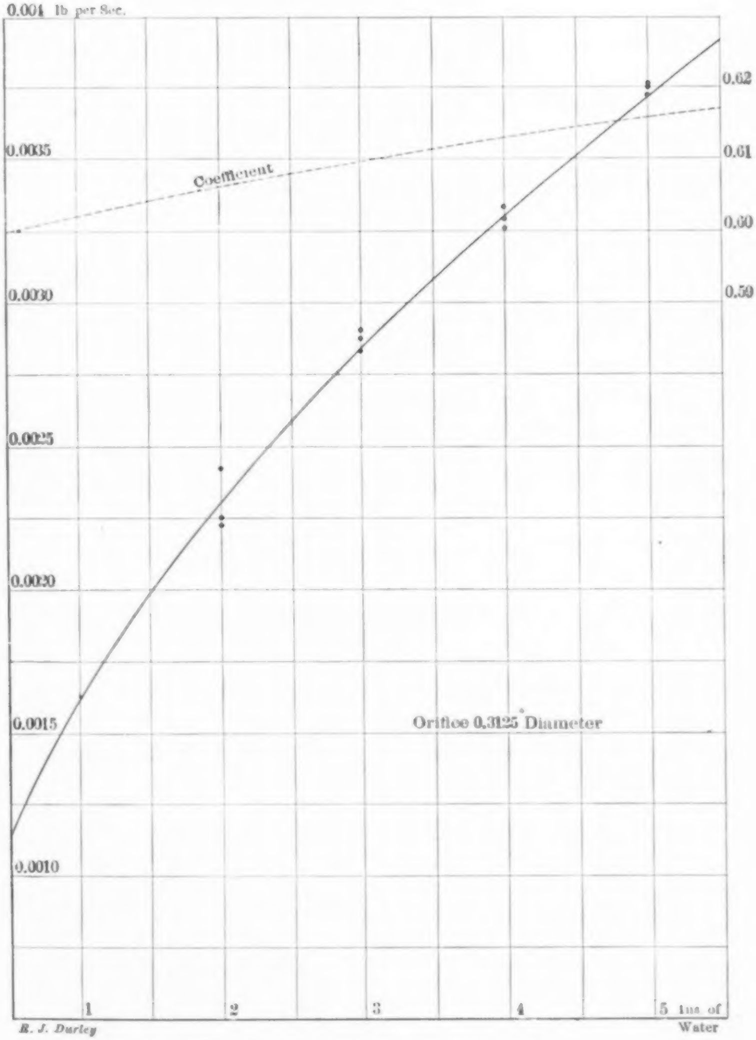


FIG. 8.—DISCHARGE FOR $\frac{5}{16}''$ ORIFICE.

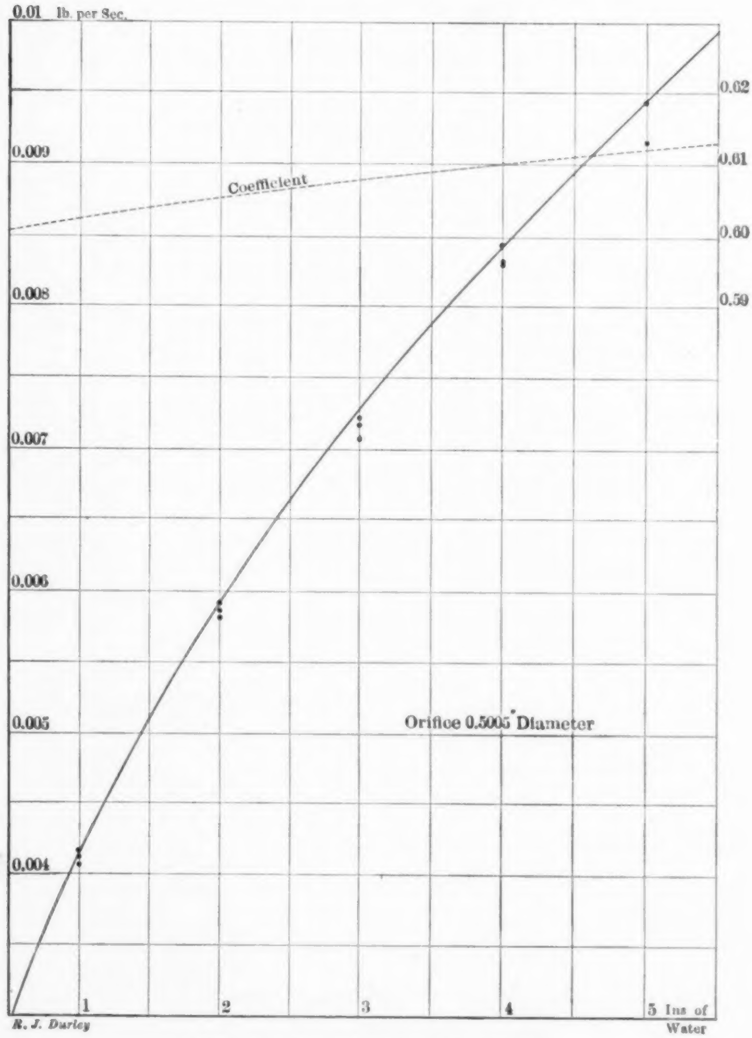


FIG. 9.—DISCHARGE FOR ½" ORIFICE.

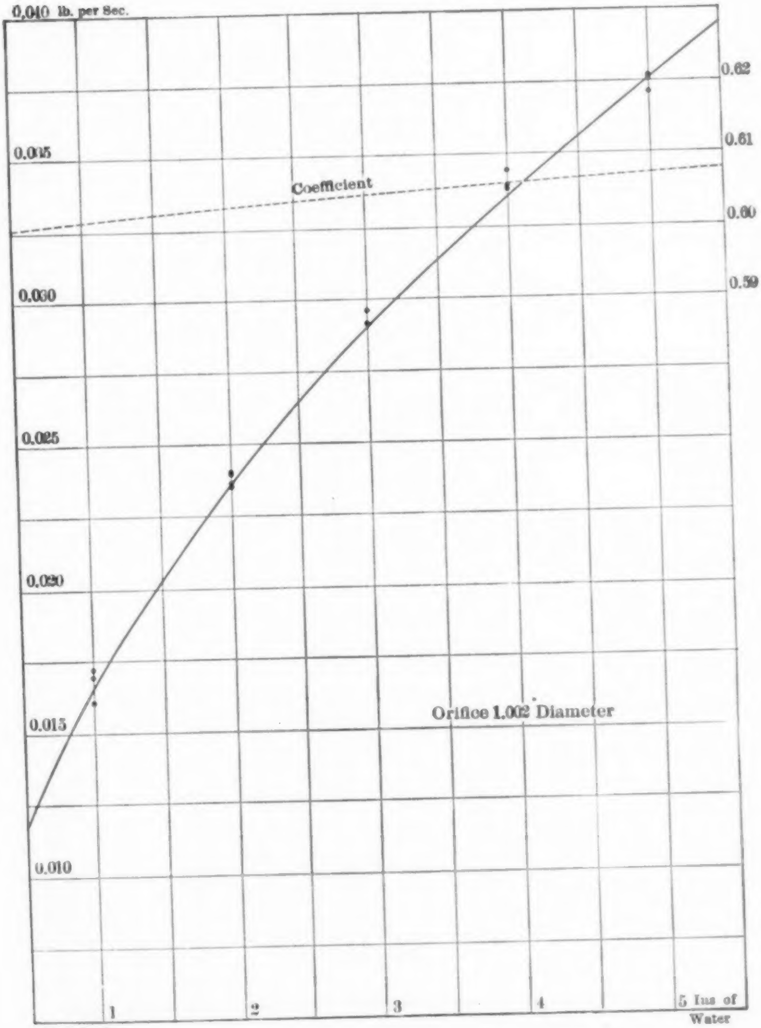


FIG. 10.—DISCHARGE FOR 1" ORIFICE.

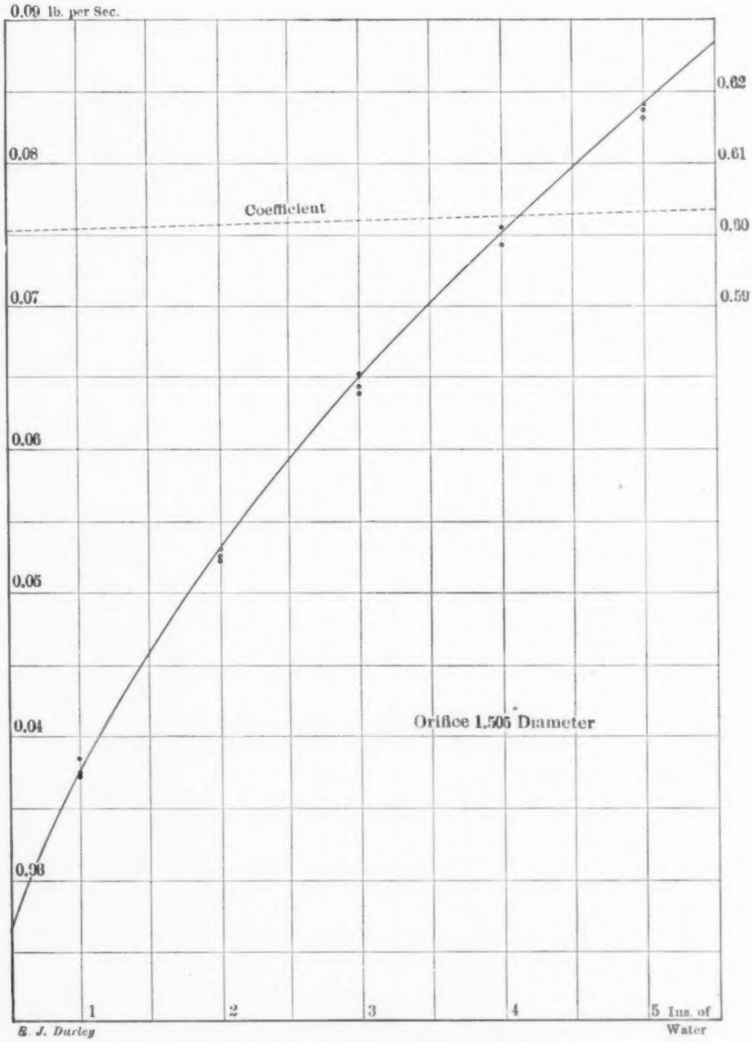


FIG. 11.—DISCHARGE FOR 1½" ORIFICE.

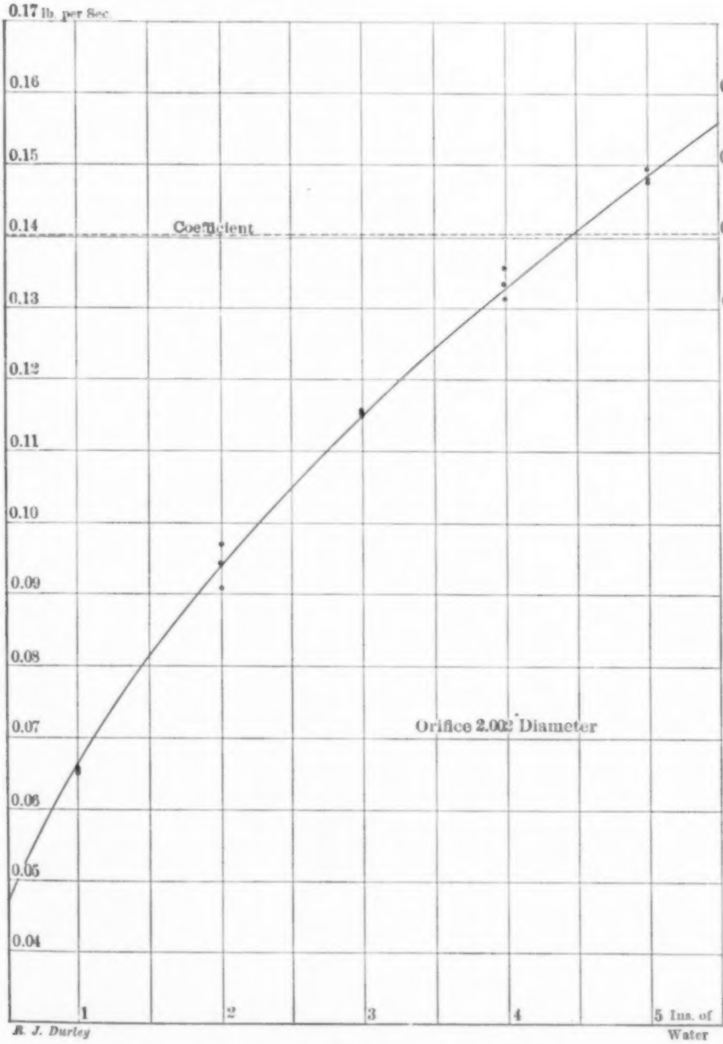


FIG. 12.—DISCHARGE FOR 2" ORIFICE.

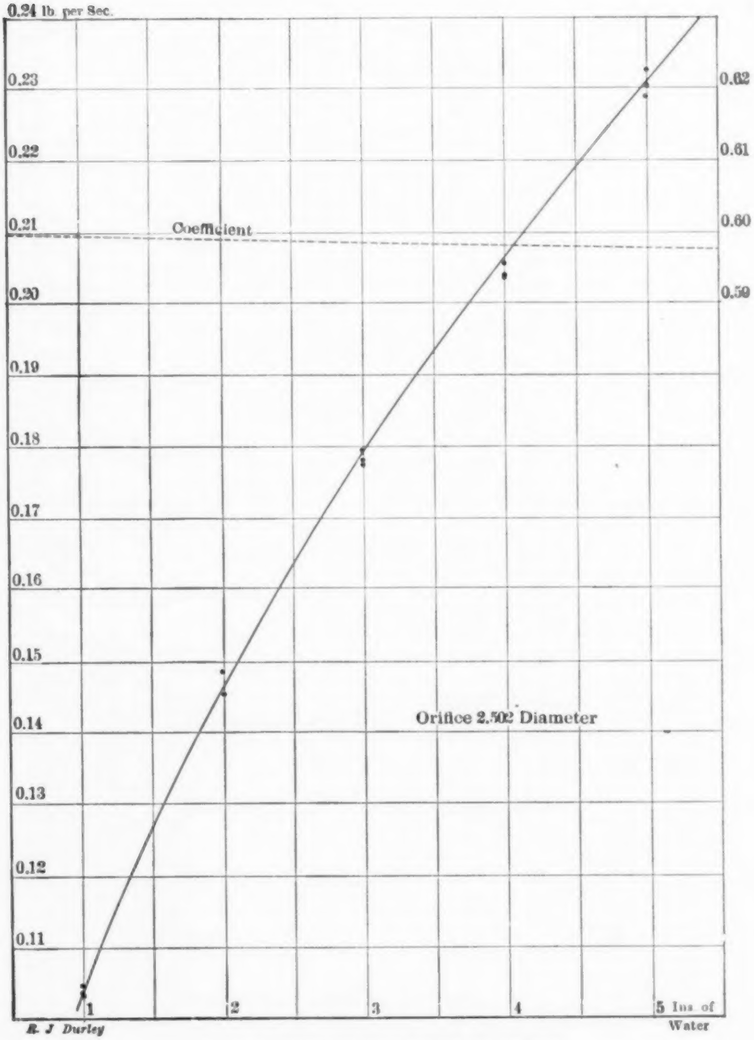


FIG. 13.—DISCHARGE FOR $2\frac{1}{2}''$ ORIFICE.

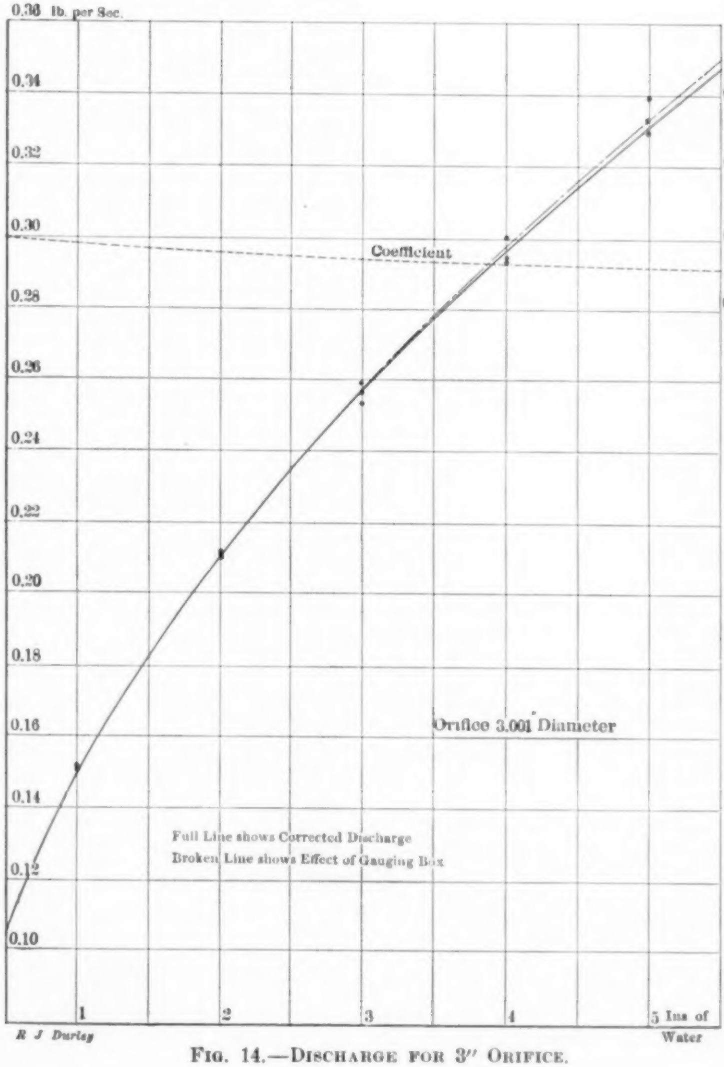


FIG. 14.—DISCHARGE FOR 3" ORIFICE.

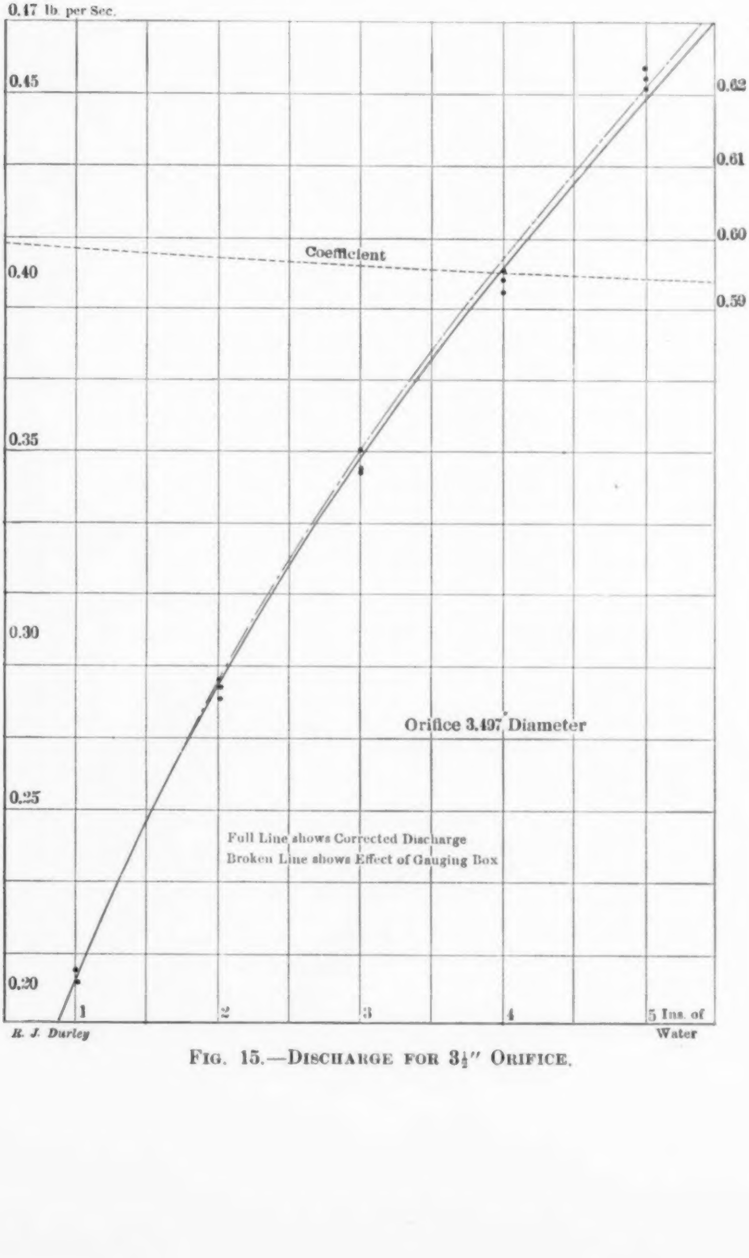


FIG. 15.—DISCHARGE FOR $3\frac{1}{2}''$ ORIFICE.

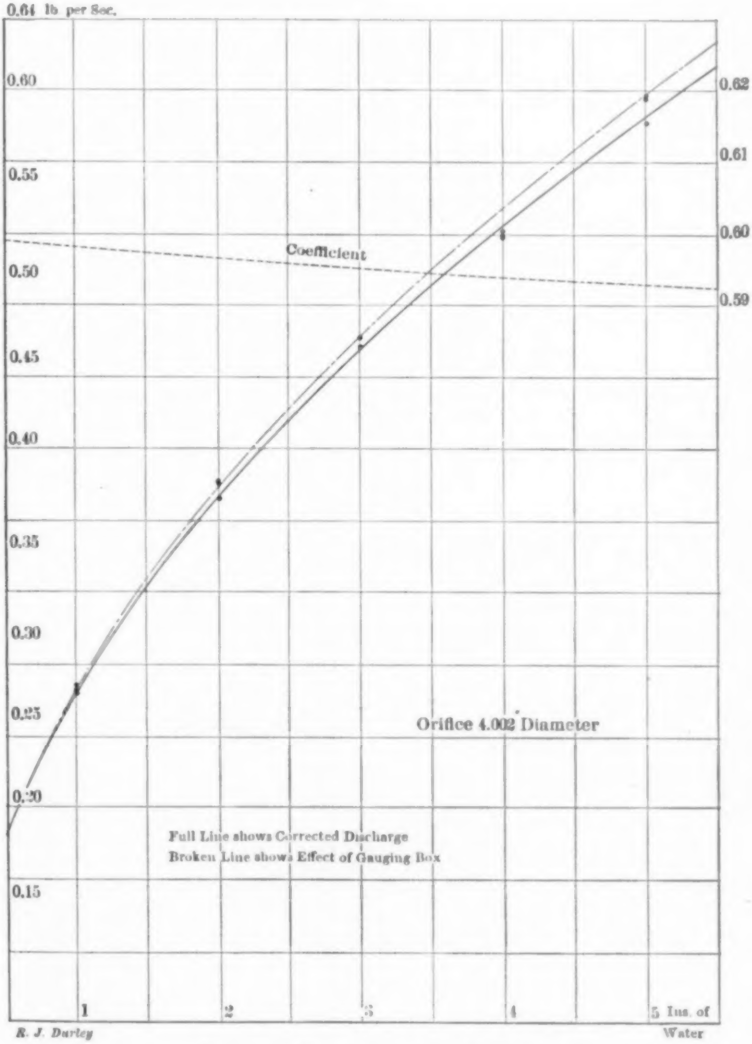


FIG. 16.—DISCHARGE FOR 4" ORIFICE.

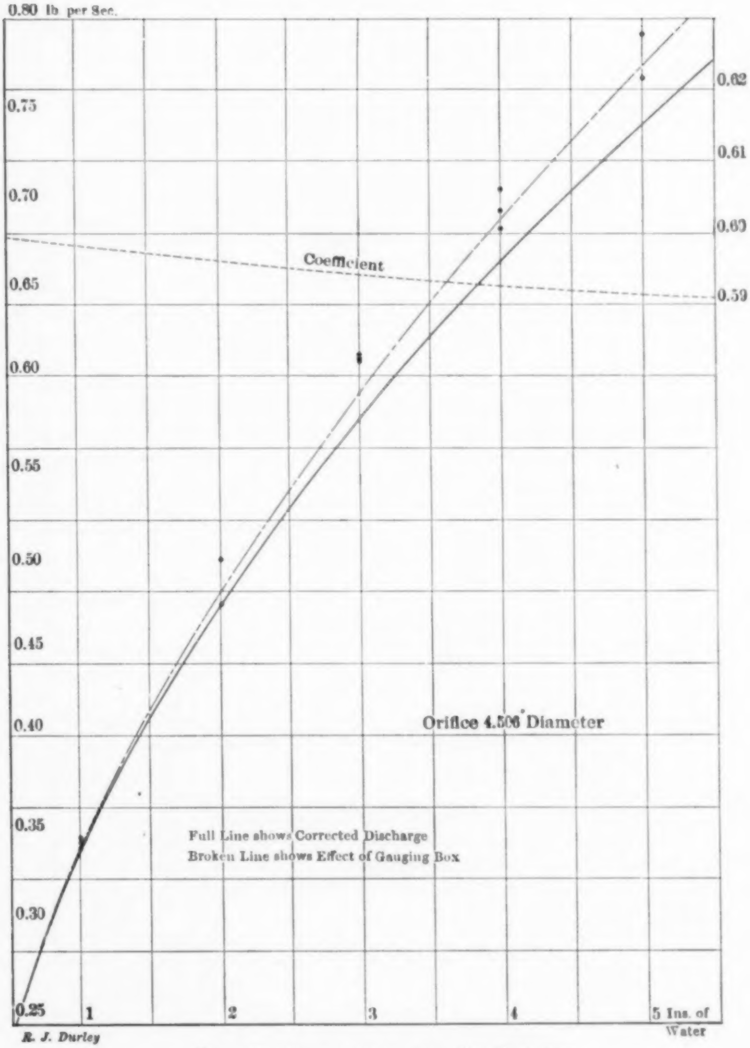


FIG. 17.—DISCHARGE FOR 4½" ORIFICE.

Conclusions and Remarks.

39. The experiments and curves indicate that:—

(1) The coefficient for small orifices increases as the head increases, but at a lesser rate the larger the orifices, till for the 2 inch orifice it is almost constant. For orifices larger than 2 inches it decreases as the head increases, and at a greater rate the larger the orifice.

(2) The coefficient decreases as the diameter of the orifice increases and at a greater rate the higher the head.

(3) The coefficient does not change appreciably with temperature (between 40 degrees and 100 degrees Fahr.).

(4) The coefficient (at heads under 6 inches) is not appreciably affected by the size of the box if the ratio of the areas of the box and orifice is at least 20:1.

40. It is hoped that the information given in this paper will enable a reasonably accurate measurement of air at low pressure to be made by the use of a comparatively simple and inexpensive gauging apparatus.

NOTE.—On derivation of equation (1).

Let W = weight of gas discharged per second in pounds.

A = area of cross section of jet in square feet.

P_1, P_2 = pressures inside and outside orifice in pounds per square foot.

V_1, V_2 = corresponding specific volumes.

γ = ratio of the specific heats K_p/K.

K = Kinetic energy of 1 lb. of gas.

u = velocity of gas at any point.

T = absolute temperature.

If heat is given to a quantity of gas under any circumstances whatever, the amount of heat is equivalent to the increase in internal energy, together with the external work done, and thus, for any small adiabatic change,

$$K_v\,dT + P\,dV = 0 \quad\text{...(a)}$$

Again if there is no frictional loss, the amount of heat given to the gas must be equivalent to the change in kinetic energy together with the change in pressure energy (head) together with the change in internal energy, so that for any small adiabatic change

$$dK + d(PV) + K_v\,dT = 0 \quad\text{...............................(b)}$$

From (a) and (b) we have

$$dK + V\,dP = 0.$$

hence in passing from conditions P_1V_1 to conditions PV we shall have

$$\int_0^u dK = \int_P^{P_1} V\,dP$$

or, since $P V^{\gamma} =$ constant, and if u is the final velocity,

$$\frac{u^2}{2g} = \frac{\gamma}{\gamma - 1}(P_1 V_1 - P_2 V_2)$$

hence $u = \sqrt{2g \frac{\gamma}{\gamma - 1}(P_1 V_1 - P_2 V_2)} = \sqrt{2g \frac{\gamma}{\gamma - 1} \cdot P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}\right]}$

and the weight of gas discharged in unit time, with frictionless adiabatic flow is

$$W = \frac{Au}{V_2} = A\sqrt{2g \frac{\gamma}{\gamma - 1} \cdot \frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma + 1}{\gamma}}\right]}$$

TABLE 1.

SOURCES OF ERROR.

Operation.	Error in Per Cent.	
	$\frac{5}{16}$-inch Orifice. 1-inch Head.	4-inch Orifice. 4-inch Head.
Allowing for leak	±3.0	±0.01
Reading pressure in reservoir	±0.2	±0.2
Reading temperature in reservoir	±0.03	±0.03
Finding volume of reservoir	±0.033	±0.033
Allowing for condensed water in reservoir	−0.025	−0.025
Measuring area of orifice	±0.7	±0.05
Taking temperature in box	±0.1	±0.1
Graduating U tube	±0.4	±0.1
Reading head from tube	±0.4	±0.1
Measuring time	±0.05	±0.06

TABLE 2.

EXPERIMENTS SHOWING EFFECTS OF TEMPERATURE OF AIR.

Diameter of orifice, 1 inch. Difference of pressure, 3 inches of water.

1	2	3	4	5	6
Expt. No.	Temperature in Gauging Box. Deg. Fahr., abs.	Actual discharge as measured. Lbs. per Second.	Corrected discharge as taken from curves, Fig. 4. Lbs. per Second.	Square Root of abs. Temperature.	Product Column 5 × Column 4.
151	516.1	0.02938	0.02910	22.72	0.6612
152	525.5	0.02901	0.02895	22.92	0.6632
153	525.9	0.02934	0.02893	22.93	0.6634
154	526.3	0.02881	0.02890	22.94	0.6630
155	537.0	0.02827	0.02860	23.17	0.6627
156	542.4	0.02864	0.02850	23.29	0.6638
157	546.8	0.02832	0.02835	23.38	0.6628
158	548.1	0.02984	0.02830	23.41	0.6626
159	549.6	0.02767	0.02825	23.45	0.6625
160	561.8	0.02637	0.02795	23.70	0.6624
161	563.3	0.03004	0.02790	23.73	0.6621
162	567.6	0.02636	0.02775	23.82	0.6610

TABLE 3.

MEAN DISCHARGE IN POUNDS PER SQUARE FOOT OF ORIFICE PER SECOND, AS FOUND FROM EXPERIMENTS.

Diameter Orifice Inches.	1-inch Head Discharge per Sq. Ft.	2-inch Head Discharge per Sq. Ft.	3-inch Head Discharge per Sq. Ft.	4-inch Head Discharge per Sq. Ft.	5-inch Head Discharge per Sq. Ft.
0.3125	3.060	4.336	5.395	6.188	7.024
0.5005	3.012	4.297	5.242	6.129	6.821
1.002	3.058	4.341	5.348	6.214	6.838
1.505	3.050	4.257	5.222	6.071	6.775
2.002	2.983	4.286	5.284	6.107	6.788
2.502	3.041	4.303	5.224	5.991	6.762
3.001	3.078	4.297	5.219	6.033	6.802
3.497	3.051	4.258	5.202	5.966	6.814
4.002	3.046	4.325	5.264	5.951	6.774
4.506	3.075	4.383	5.508	6.260	7.028

TABLE 4.

COEFFICIENTS OF DISCHARGE FOR VARIOUS HEADS AND DIAMETERS OF ORIFICE.

Diameter of Orifice Inches.	1-inch Head.	2-inch Head.	3-inch Head.	4-inch Head.	5-inch Head.
$\frac{5}{16}$	0.603	0.606	0.610	0.613	0.616
$\frac{1}{2}$	0.602	0.605	0.608	0.610	0.613
1	0.601	0.603	0.605	0.606	0.607
$1\frac{1}{2}$	0.601	0.601	0.602	0.603	0.603
2	0.600	0.600	0.600	0.600	0.600
$2\frac{1}{2}$	0.599	0.599	0.599	0.598	0.598
3	0.599	0.598	0.597	0.596	0.596
$3\frac{1}{2}$	0.599	0.597	0.596	0.595	0.594
4	0.598	0.597	0.595	0.594	0.593
$4\frac{1}{2}$	0.598	0.596	0.594	0.593	0.592

Weight of air discharged per second $= 0.6299\, Cd^2 \sqrt{\frac{i}{T}}$ lbs. (Barometer at 30 inches.)

TABLE 5.

CORRECTED ACTUAL DISCHARGE IN POUNDS PER SECOND AT 60 DEGREES FAHR. AND 14.7 POUNDS BAROMETRIC PRESSURE FOR CIRCULAR ORIFICES IN PLATE 0.057 INCHES THICK.

Head Inches.	DIAMETER OF ORIFICE IN INCHES.										
	0.3125	0.500	1.000	1.500	2.000	2.500	3.000	3.500	4.000	4.500	5.000
$\frac{1}{2}$	0.00114	0.00293	0.0117	0.0263	0.0468	0.0732	0.105	0.143	0.187	0.237	0.292
1	0.00162	0.00416	0.0166	0.0373	0.0663	0.103	0.149	0.202	0.264	0.334	0.413
$1\frac{1}{2}$	0.00199	0.00510	0.0203	0.0457	0.0811	0.127	0.182	0.248	0.323	0.409	0.505
2	0.00231	0.00590	0.0235	0.0528	0.0937	0.146	0.210	0.285	0.373	0.471	0.582
$2\frac{1}{2}$	0.00259	0.00662	0.0263	0.0591	0.105	0.163	0.235	0.319	0.416	0.526	0.649
3	0.00285	0.00726	0.0289	0.0648	0.115	0.179	0.257	0.349	0.455	0.575	0.710
$3\frac{1}{2}$	0.00308	0.00786	0.0312	0.0700	0.124	0.193	0.277	0.377	0.491	0.621	0.766
4	0.00330	0.00842	0.0334	0.0749	0.133	0.206	0.296	0.402	0.525	0.663	0.817
$4\frac{1}{2}$	0.00351	0.00895	0.0355	0.0794	0.141	0.219	0.314	0.426	0.556	0.702	0.865
5	0.00371	0.00945	0.0375	0.0838	0.148	0.231	0.331	0.449	0.586	0.739	0.912
$5\frac{1}{2}$	0.00390	0.00993	0.0393	0.0879	0.155	0.242	0.347	0.471	0.613	0.774	0.953
6	0.00408	0.01049	0.0411	0.0918	0.162	0.252	0.362	0.492	0.640	0.808	0.995

TABLE 6.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir. Abs. Pressures. Initial.	Reservoir. Abs. Pressures. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
1	5/16	1	1,260	88.8	83.4	520.1	520.0	518.6	.002033	.001628
2	"	1	3,900	95.4	78.7	520.1	520.0	518.5	.002040	.001631
3	"	2	1,200	71.8	64.8	519.9	519.8	520.0	.002758	.002430
4	"	2	3,600	78.2	58.7	520.0	519.8	520.0	.002592	.002264
5	"	2	508	78.2	19.4	542.8	531.5	534.0	.002463	.002236
6	"	3	1,200	57.6	49.7	520.0	519.8	521.0	.003149	.002879
7	"	3	1,200	49.7	41.8	519.8	519.8	521.3	.003152	.002912
8	"	3	3,600	57.6	34.5	520.0	519.8	521.3	.003072	.002832
9	"	4	1,800	92.14	78.22	520.9	520.6	521.4	.003677	.003292
10	"	4	1,560	78.22	66.31	520.6	520.5	520.5	.003637	.003296
11	"	4	4,080	92.14	60.79	520.9	520.4	520.2	.003665	.003262
12	"	4	373	92.5	27.6	542.3	530.6	531.2	.003699	.003333
13	"	5	2,400	59.9	39.8	520.4	519.9	521.5	.003998	.003748
14	"	5	2,700	101.18	77.39	522.1	520.1	522.2	.004157	.003751
15	"	5	3,900	101.18	67.27	522.1	520.5	522.3	.004109	.003723

TABLE 7.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial. Secs.	Reservoir. Abs. Pressures. Initial.	Reservoir. Abs. Pressures. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
16	1/2	1	3,398	96.8	63.7	529.2	523.4	524.5	.004482	.004123
17	1/2	1	1,153	96.8	85.0	529.2	524.0	524.4	.004468	.004067
18	1/2	1	2,245	85.0	63.7	524.0	523.4	524.5	.004489	.004155
19	1/2	2	2,427	96.8	64.7	524.9	522.6	524.4	.006227	.005869
20	1/2	2	1,276	96.8	79.9	524.9	523.2	524.4	.006200	.005822
21	1/2	2	1,151	79.9	64.7	523.2	522.6	524.1	.006247	.005924
22	1/2	3	2,992	103.0	54.6	528.8	523.1	524.9	.007518	.007177
23	1/2	3	1,081	103.0	85.0	528.8	524.1	524.5	.007482	.007078
24	1/2	3	1,911	85.0	54.6	524.1	523.1	525.0	.007541	.007236
25	1/2	4	2,126	93.8	54.6	525.3	522.9	523.4	.008674	.008346
26	1/2	4	1,036	93.8	74.7	525.3	523.5	523.0	.008600	.008322
27	1/2	4	1,090	74.7	54.6	523.5	522.9	523.8	.008744	.008457
28	1/2	5	981	94.8	74.7	526.4	524.6	518.9	.009561	.009185
29	1/2	5	533	65.7	54.6	524.3	522.8	521.1	.009757	.009457

TABLE 8.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of Trial. Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
30	1	1	412	87.0	72.5	522.6	521.0	519.1	.01643	.01603
31	1	1	833.67	94.17	63.39	522.1	521.4	513.8	.01752	.01699
32	1	1	870.14	93.51	60.90	521.1	518.8	515.5	.01773	.01723
33	1	2	1,157	92.1	32.6	529.5	518.0	520.0	.02389	.02357
34	1	2	587	62.5	32.6	519.6	518.0	520.2	.02432	.02406
35	1	2	1,147	92.1	32.6	533.1	518.2	520.4	.02384	.02353
36	1	2	587	62.5	32.6	520.0	518.2	520.1	.02424	.02398
37	1	3	478	60.5	30.6	520.1	518.6	522.1	.02984	.02963
38	1	3	748	83.9	36.5	524.5	520.6	521.7	.02990	.02963
39	1	3	1,020	91.3	27.5	522.6	518.2	516.1	.02967	.02925
40	1	4	676	94.8	44.8	523.4	517.0	520.1	.03477	.03440
41	1	4	658	94.8	44.8	535.8	517.4	524.6	.03415	.03393
42	1	4	776	91.8	34.6	528.8	515.4	519.8	.03414	.03379
43	1	5	593	94.8	44.8	538.8	518.5	518.5	.03758	.03715
44	1	5	588	94.8	44.8	537.9	518.3	518.8	.03798	.03756
45	1	5	587	94.8	44.8	538.0	518.3	518.7	.03806	.03764

TABLE 9.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of Trial. Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
46	1½	1	572	91.8	44.8	524.1	519.5	519.0	.03870	.03839
47	1½	1	592	94.8	44.8	538.3	518.8	519.3	.03772	.03732
48	1½	1	586	94.8	44.8	541.3	519.5	519.7	.03773	.03734
49	1½	2	417	94.8	44.8	540.8	515.5	518.0	.05274	.05224
50	1½	2	600	94.7	26.6	523.4	517.0	513.4	.05368	.05301
51	1½	2	600	94.3	26.8	523.2	515.7	514.5	.05318	.05256
52	1½	3	373	96.8	44.8	523.7	512.7	516.6	.06500	.06440
53	1½	3	319	89.8	44.8	523.4	512.8	516.7	.06570	.06514
54	1½	3	339	92.8	44.8	529.4	513.1	516.1	.06459	.06399
55	1½	4	286	91.8	44.8	523.8	512.1	516.2	.07632	.07564
56	1½	4	306	94.8	44.8	526.9	512.2	514.9	.07518	.07443
57	1½	5	273	94.8	44.8	524.7	511.1	517.1	.08480	.08415
58	1½	5	267	94.8	44.8	532.0	511.6	515.2	.08445	.08326
59	1½	5	256	91.8	44.8	527.7	512.1	514.8	.08401	.08321

TABLE 10.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
60	2	1	347.2	92.1	42.5	524.6	510.1	511.0	.06613	.06521
61	2	1	332.0	87.0	37.5	539.8	510.6	513.2	.06579	.06503
62	2	1	335.2	87.0	37.5	536.4	512.8	513.7	.06613	.06539
63	2	2	240.5	82.0	32.6	537.2	511.4	513.0	.09191	.09097
64	2	2	242.0	87.0	37.5	525.5	511.5	517.1	.09479	.09416
65	2	2	232.7	94.8	44.8	533.5	511.3	521.2	.09632	.09600
66	2	3	240.0	95.8	36.0	526.2	516.4	512.6	.1165	.1153
67	2	3	240.0	94.8	35.3	525.6	522.3	513.3	.1168	.1157
68	2	3	240.0	91.8	32.4	525.5	520.1	514.7	.1165	.1155
69	2	3	210.0	93.1	41.1	526.2	524.1	513.6	.1167	.1156
70	2	4	150.0	89.3	45.7	525.5	523.9	514.4	.1371	.1358
71	2	4	150.0	88.8	45.9	526.0	521.9	514.9	.1343	.1333
72	2	4	150.0	96.6	53.8	527.3	519.7	515.0	.1322	.1312
73	2	5	119.31	92.2	54.3	523.3	521.4	517.5	.1502	.1494
74	2	5	113.93	90.6	54.6	524.6	523.0	516.1	.1491	.1481
75	2	5	132.18	96.8	55.1	524.5	522.2	514.3	.1489	.1476

TABLE 11.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
76	2½	1	299.8	92.8	26.5	523.7	515.6	516.1	.1042	.1034
77	2½	1	300.0	93.8	27.4	523.2	515.0	513.3	.1046	.1036
78	2½	1	299.5	92.9	25.9	522.8	516.1	511.9	.1059	.1047
79	2½	1	300.0	93.3	26.6	524.9	514.0	511.1	.1046	.1034
80	2½	2	139.0	90.6	48.1	516.7	515.5	510.7	.1470	.1453
81	2½	2	150.0	90.38	42.92	521.1	515.8	513.6	.1499	.1486
82	2½	3	178.2	92.1	22.3	532.8	506.8	509.8	.1800	.1780
83	2½	3	105.0	87.5	48.4	513.8	515.3	512.8	.1810	.1793
84	2½	3	110.0	89.93	48.15	522.2	518.1	512.3	.1793	.1776
85	2½	4	126.6	92.1	32.6	545.8	509.2	511.5	.2059	.2038
86	2½	4	99.0	90.96	47.33	521.5	514.6	514.7	.2074	.2059
87	2½	4	98.66	86.0	42.8	523.4	516.9	512.9	.2058	.2039
88	2½	5	111.4	89.0	32.6	526.0	507.7	513.8	.2346	.2328
89	2½	5	114.8	92.1	32.6	537.1	509.7	509.1	.2335	.2306
90	2½	5	115.8	96.7	36.5	537.3	508.7	505.8	.2327	.2290

TABLE 12.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir.				Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
				Abs. Pressure.		Abs. Temperature.				
				Initial.	Final.	Initial.	Final.			
91	3	1	150.17	98.3	49.6	524.4	521.2	513.2	.1530	.1513
92	3	1	148.32	96.5	48.5	524.4	522.8	513.8	.1532	.1513
93	3	1	153.63	95.9	45.8	526.6	523.0	512.1	.1531	.1509
94	3	2	86.21	92.1	53.1	523.9	523.6	513.5	.2147	.2111
85	3	2	93.72	93.0	50.8	519.0	517.4	512.6	.2138	.2107
96	3	2	74.95	93.2	59.3	520.0	519.3	509.9	.2161	.2114
97	3	3	67.32	99.1	62.0	519.9	517.6	509.3	.2613	.2567
98	3	3	73.20	94.0	53.6	520.3	519.5	512.5	.2638	.2591
99	3	3	44.45	99.2	73.7	521.8	511.4	509.5	.2569	.2533
100	3	4	47.92	94.8	63.0	519.8	508.8	509.3	.3043	.3006
101	3	4	45.54	97.1	68.3	519.2	517.0	511.5	.2995	.2944
102	3	4	55.15	100.7	65.6	521.8	517.5	509.7	.2986	.2941
103	3	5	42.97	90.0	59.3	519.3	516.9	510.4	.3388	.3336
104	3	5	50.23	98.0	62.4	520.1	518.8	505.7	.3373	.3292
105	3	5	43.27	94.6	63.1	518.8	518.6	506.3	.3490	.3397

TABLE 13.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir.				Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
				Abs. Pressure.		Abs. Temperature.				
				Initial.	Final.	Initial.	Final.			
106	3½	1	72.10	99.5	67.6	520.2	518.7	505.5	.2105	.2051
107	3½	1	67.35	98.5	69.1	519.6	517.6	505.7	.2069	.2020
108	3½	2	52.2	100.4	68.7	520.4	518.9	505.2	.2889	.2810
109	3½	2	68.78	99.1	56.8	520.1	518.1	503.7	.2926	.2862
110	3½	2	54.59	98.3	64.9	520.3	519.7	505.1	.2926	.2847
111	3½	3	41.04	96.23	65.60	519.8	517.9	507.2	.3541	.3458
112	3½	3	69.2	89.7	38.3	522.5	512.1	509.2	.3481	.3442
113	3½	3	75.0	89.0	33.2	520.0	515.5	511.2	.3543	.3509
114	3½	4	34.04	98.33	69.28	521.5	519.6	505.7	.4042	.3945
115	3½	4	60.5	86.9	34.9	520.3	515.7	503.5	.4084	.4012
116	3½	4	68.5	90.9	32.4	521.0	511.8	506.4	.4039	.3984
117	3½	5	29.35	95.15	66.21	519.2	513.1	507.0	.4595	.4518
118	3½	5	30.12	95.41	65.94	520.3	519.2	506.6	.4650	.4545
119	3½	5	31.22	97.69	66.92	520.7	520.1	505.1	.4702	.4572

TABLE 14.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir.				Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
				Abs. Pressure.		Abs. Temperature.				
				Initial.	Final.	Initial.	Final.			
120	4	1	54.89	98.00	66.57	520.6	518.9	507.8	.2715	.2657
121	4	1	48.69	96.95	68.56	522.0	519.3	506.7	.2745	.2684
122	4	1	47.46	97.30	70.03	521.6	518.3	506.6	.2696	.2642
123	4	2	36.08	96.31	67.24	521.1	520.0	505.4	.3820	.3722
124	4	2	38.60	97.39	65.60	521.6	518.9	505.2	.3887	.3805
125	4	2	40.6	97.15	63.73	520.8	517.0	507.7	.3880	.3811
126	4	3	53.5	89.6	37.1	523.7	519.7	508.3	.4636	.4578
127	4	3	54.6	91.8	37.2	524.3	514.5	506.8	.4684	.4620
128	4	4	22.69	94.65	69.07	521.4	518.6	504.8	.5319	.5189
129	4	4	21.66	95.49	70.63	521.4	517.0	503.6	.5330	.5191
130	4	4	42.5	91.2	43.0	524.3	514.8	506.0	.5298	.5221
131	4	5	19.91	96.55	71.33	520.5	519.1	503.7	.6125	.5970
132	4	5	20.32	97.35	71.17	519.8	518.7	503.4	.6135	.5967
133	4	5	20.68	93.94	67.75	520.1	516.4	504.6	.5943	.5819

TABLE 15.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 60° F. and 14.7 lbs.
134	4½	1	38.00	97.85	69.83	521.0	519.4	504.8	.3489	.3406
135	4½	1	47.83	102.48	66.96	525.5	518.9	507.3	.3429	.3373
136	4½	1	40.25	97.13	67.37	520.9	518.9	506.2	.3505	.3438
137	4½	2	23.57	99.0	74.6	522.1	518.9	506.2	.4834	.4727
138	4½	2	25.89	93.50	65.77	515.9	512.7	505.4	.5086	.4984
139	4½	3	23.79	96.31	65.18	515.7	512.9	503.8	.6233	.6085
140	4½	3	25.65	97.74	64.40	515.4	516.0	503.4	.6295	.6099
141	4½	3	23.82	95.77	64.77	515.0	515.8	503.4	.6308	.6119
42	4½	4	21.46	96.82	64.62	515.3	515.4	503.5	.7247	.7044
143	4½	4	21.31	94.35	63.30	515.2	515.8	503.4	.7052	.6832
144	4½	4	37.00	94.47	39.95	515.1	514.5	498.9	.7112	.6925
145	4½	5	29.06	93.40	44.37	515.9	515.0	498.2	.8121	.7902
146	4½	5	29.71	93.43	44.80	515.9	516.3	496.5	.7893	.7664
147	3	1	93.85	96.61	65.90	520.2	517.9	509.1	.1548	.1512
148	3	1	87.81	97.21	68.31	521.1	519.0	508.0	.1556	.1515
149	3	5	36 59	92.35	63.41	520.2	518.1	504.7	.3767	.3675
150	3	5	34.74	91.02	63.41	520.3	519.8	504.4	.3796	.3679

TABLE 16.

EXPERIMENTS ON EFFECT OF VARIATIONS OF TEMPERATURE.

No. of trial.	Diam. of Orifice ins.	Head ins. water.	Duration of trial Secs.	Reservoir. Abs. Pressure. Initial.	Reservoir. Abs. Pressure. Final.	Reservoir. Abs. Temperature. Initial.	Reservoir. Abs. Temperature. Final.	Box Temp. Abs.	Actual Discharge Lbs. Sec.	Corrected actual discharge at 14.7 lbs.
151	1	3	1,020	91.3	27.5	522.6	518.2	516.1	.02968	.02938
152	1	3	600.0	93.9	55.9	525.4	517.9	525.5	.02936	.02901
153	1	3	600.0	87.2	49.1	526.3	517.4	525.9	.02936	.02934
154	1	3	600.0	94.2	56.1	527.5	517.8	526.3	.02916	.02881
155	1	3	600.0	90.6	52.6	531.7	519.1	537.0	.02861	.02827
156	1	3	600.0	89.3	52.2	524.0	519.3	542.4	.02898	.02864
157	1	3	600.0	87.6	50.5	525.6	518.0	546.8	.02865	.02832
158	1	3	600.0	92.1	52.0	531.8	517.7	548.1	.03018	.02984
159	1	3	599.5	86.3	50.2	524.8	518.4	549.6	.02803	.02767
160	1	3	599.0	90.6	55.1	533.5	522.4	561.8	.02675	.02637
161	1	3	600.0	91.8	51.5	531.3	517.3	563.3	.03038	.03004
162	1	3	599.4	80.0	46.0	523.7	520.1	567.6	.02671	.02636

DISCUSSION.

Mr. Walter B. Snow.—Last month, at the meeting of the Society of Naval Architects and Marine Engineers, D. W. Taylor, in his paper on "Experiments with Ventilating Fans and Pipes," proposed a method of measuring air by means of a series of Pitot tubes. Great care and precision were evident in the conduct of the experiments upon which this paper was based.

The paper which we have under consideration to-day, although proposing a different method, manifests the same thought and

care as to details. Its value is greatly increased by the clearness with which the writer illustrates his method of test and calculations, but it appears to lack any explanation as to the form of the tube connection inside the box.

Inasmuch as the author assumes that the velocity of approach of the air has some influence on the air deliveries with the larger orifices, it also seems reasonable to assume that the form and location of the U tube connection inside the box might have an appreciable effect upon the indicated pressure. It is certainly true that even with moderate velocities the aspiration effect of the air flowing squarely across the end of a tube will be appreciable and erratic, as was pointed out in the former paper. If the end of the tube be curved, the total pressure will be indicated only when its mouth directly faces the current of air.

It is interesting to note that for the two-inch orifices the author finds the coefficient for the formula

$$\text{Pounds per second} = 0.6299\ Cd^2\sqrt{\frac{i}{T}}$$

to be a constant and equal to 0.60. This being the case, I would suggest that a very convenient form of measuring device might be made of a thin plate with a number of two-inch orifices so arranged that they could be readily opened and closed. Varying quantities of air might thus be measured while using a constant coefficient for different heads. The only variables requiring to be noted would be the pressure of the air, its temperature and the number of two-inch orifices which were open. The formula would then resolve itself into

$$\text{Pounds per second} = 1.512\ N\sqrt{\frac{i}{T}}$$

where N = number of two-inch openings.

In experiments of great precision the humidity of the air should be taken into account as well as its temperature and pressure. Under ordinary atmospheric conditions the humidity makes a difference of only one-half of one per cent. in the density, but for unusual conditions it is certainly worthy of notice.

Mr. Albert Kingsbury.—I am much interested in this paper, and I think we all should be grateful for the results which the author has presented to us.

I have at times been desirous of measuring the volume of discharge of fans, and have been to some extent deterred from under-

taking the work because of the very great difficulty that is involved in securing reasonably exact results.

It seems to me that the results of these experiments afford an outlet in this direction. We may avoid a large amount of experimental work by utilizing the results of the experiments.

I wish to ask the author if he thinks the coefficient of outflow is affected to any extent by the thickness of the plate at the orifice. The orifices which were used were all straight bored; that is, the edge of the plate was not beveled. The orifice is, therefore, really a short tube, the length of the tube being the thickness of the plate, about 1-20 inch. My question is whether beveling of the edge of plate would appreciably affect the coefficient of discharge.

Mr. H. H. Suplee.—I should like to ask Professor Durley if he has made any attempt to obtain a constant coefficient by the use of fractional exponents, as has been done by Tutton and others with the Chézy formula for the flow of water. With the original form of the Chézy formula, $v = C\sqrt{rs}$, the coefficient C was found to be far from constant. By changing the formula to the form: $v = C\,r^m\,s^n$ and determining separate values for the exponents m and n it has been found possible to maintain the coefficient C practically constant for the whole range of velocities, changing it only for surfaces of different kinds. In practice it has been found that the formula $v = C\,r^{\frac{2}{3}}\,s^{\frac{1}{2}}$ has a coefficient which varies only with the changes in the nature of the surface of the pipe or conduit and not with the velocities. It seems as if a similar method applied to the valuable experiments of Professor Durley would permit of the computation, by graphical methods, using logarithmically ruled paper, or otherwise, of exponents which would cause the coefficient to remain constant for the flow of air through orifices.

Mr. Sanford A. Moss.—To the list of experiments on the subject which Professor Durley gives could be added a reference to "An Experimental Determination of the Coefficient of Discharge of Air," *American Machinist,* August 10, 1905, Volume 28, page 193. In this article a set of experiments very similar to those of Professor Durley is described, and the theory developed. The work was done with an orifice with a well-rounded approach, with pressures from four to eight pounds per square inch above atmosphere. The result obtained was a value for the coefficient of discharge of 0.942. The method used was very similar to that used by Professor Durley.

Prof. D. S. Jacobus.—My experience with the flow of steam shows the effect of one orifice on the other, where a number are placed near each other in a plate. In an experiment, the flow of steam through a single orifice $\frac{3}{8}$ inch in diameter in a plate placed between flanges in a 2-inch pipe was compared with that obtained with six orifices in the same size plate. The diameter of the orifices was $\frac{3}{8}$ inch, and the pressure on one side of the plate was maintained at 147 pounds above the atmosphere, and on the other side at 105 pounds above the atmosphere. The flow per orifice was about 14 per cent. greater with the plate having the six orifices than with the plate having a single orifice. The pressure on the low-pressure side of the orifices was measured at some little distance from the orifice plate, in order to avoid a reduction of pressure which was found to exist near the plate.

As an eddy or suction action at an orifice may affect the result to so great an extent, is it not possible that the size of the discharge end of the gauging box used in Prof. Durley's test may have affected the results? If, for example, a shield had been placed in the plane of the orifice at the discharge end of the box so that the orifice would be small in proportion to the plane surface in which it was placed, might not the results have been somewhat different? If such is the case, it would be well to call attention to the fact in order to make sure that the results are not wrongly interpreted.

*Prof. R. J. Durley.**—In replying to the gentlemen who have been kind enough to discuss this paper, I should like to say, in connection with Mr. Snow's remarks, that I have done a good deal of work in endeavoring to measure the discharge of air by means of Pitot tubes, and the results have not been encouraging. I have not been able to make such measurements in ordinary air ducts with any large degree of success, and it was really for that very reason that I undertook the work which is described in this paper.

It seems to me that the trouble in endeavoring to measure air by means of Pitot tubes arises from two causes. In the first place, the velocity head is often very small—at least, when dealing with small differences of pressure, and secondly, it is extremely difficult to place the Pitot tubes in such positions that they will be in a steady, uniform stream of air, free from eddies.

Mr. Snow raised a point about the location of the pressure-

* Author's closure under the Rules.

gauging tube in the gauging box, and I may say that we tried many positions for this tube, taking care in each case that the opening on the inner surface of the box was flush with the wood. We could not find any difference in the indications unless the tube was placed too close to the inlet or outlet. The water gauge was intended to show the static pressure in the box, and I think really that it did so.

In regard to the question of the velocity of approach, I think for practical purposes it is much better to make a box big enough in cross-section, so as to make the effect of that velocity negligible, than to endeavor to allow for the velocity of approach.

Mr. Snow's suggestion as to the use of a number of two-inch orifices would no doubt be correct as long as care is taken not to place the orifices too close to each other in the perforated plates. In using an orifice in a thin plate of this kind, the influence of the stream is found to some extent on the inner side of the plate, and it is probable that the jets on the outside would also affect each other in the manner described by Professor Jacobus.

Mr. Kingsbury discussed the variation of the coefficient of discharge with various sizes of orifice. I think that his explanation is the correct one, and that such variation is due to the fact that these orifices were really short lengths of parallel tube.

When trying bevel-edge orifices, I found there was considerable difficulty in insuring that all had exactly the same taper, and that the edges were uniformly thin, and that the openings were truly circular. For these reasons it seemed better to use orifices of the form described in the paper, and I am unable to say what the coefficient of discharge would be for bevel-edge orifices.

Mr. Suplee's suggestion as to the use of fractional exponents in the expression for the discharge is a very interesting one. It seems questionable, however, whether the resulting logarithmic diagrams would prove more convenient for practical use than the curves and tables of the paper.

No. 1099.*

THE PRESSURE DROP THROUGH POPPET VALVES.

BY CHARLES EDWARD LUCKE, NEW YORK, N. Y.
(Associate Member of the Society.)

1. All gas engines, most air compressors and ice machines and many steam engines, are fitted with poppet valves to control the fluid-flow into or from the cylinders. These machines operate at widely different piston speeds and use fluids of widely differing densities with consequent varying velocities through the valves. The pressure drop for a given rate of flow, however, is dependent upon the fluid density, as well as its velocity, the form of opening, the form of approach and exit, the character of the flow, whether steady or intermittent, the inertia of the fluid and the friction of the passages. Although pressure drop through the valve is necessary to secure any flow whatever, such pressure drop is detrimental to the machine, causing, as it does, undesirable piston resistance and reduced volumetric efficiency of the cylinder. Pressure drop is reduced to a minimum in designing by allowing proper valve area to minimize fluid velocity, but too large a poppet valve for a given gas, or too large a lift for any given valve, introduces features of design equally as bad as too small a valve opening. The designer is thus forced to follow a middle course admitting some pressure drop through the valve, but not reducing fluid velocity so low as to require valves of excessively large diameter or excessively large lift. To design such a valve there should be available experimental data on the relation between pressure drop and the conditions tending to produce it, because it cannot be calculated on theoretic grounds, but on investigation it will be found that such information is unavailable.

2. The tests reported here had for their object the determination of some such experimental data on the relation between pressure

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* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

drop through a poppet valve and the condition of flow, form of opening, etc., that give rise to the pressure drop observed. For any given velocity of flow the pressure drop will be greatest and most easily measured with a less dense fluid; for this reason atmospheric air was used. The gas engines in the experimental laboratories of Columbia University have poppet valves with flat and conical sets and are operated both automatically and by cams. These valves were so arranged for these tests as to allow measured quantities of air to flow both steadily and intermittently with various valve lifts and in both directions through the valves. The valve lift was measured by one special attachment for the tests made on the engines themselves and another for the tests on the detached valves with steady flow. The air was measured by a large Westinghouse meter and all pressure drops were measured in inches of water on a manometer. The experimental work was carried out as a thesis problem by two students of the graduating class, Mr. R. M. Strong and Mr. F. W. Hollman, to whom proper credit should be given.

3. The flat, seat valve used was one from a Daimler engine, 1.58 inch inside diameter, arranged as shown in Fig. 1. For

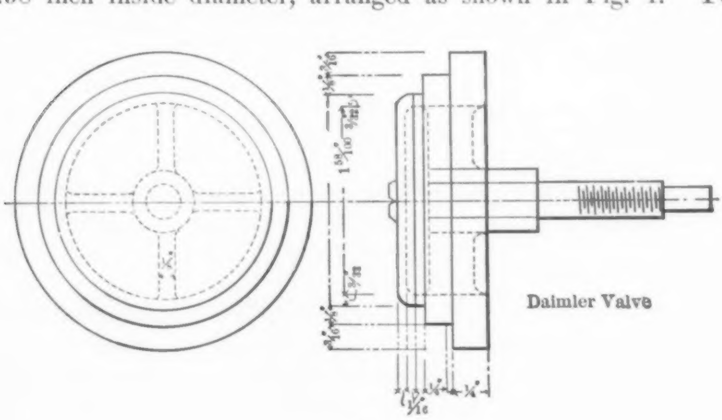


FIG. 1.

steady flow runs it was held open by small copper wedges carefully ground to size, three being used at a time, inserted at the outside edge of the valve so that the valve opening which is measured by the inside edge of the valve would not be restricted by the wedges. These wedges were of such size as to allow of four different lifts, 0.05, 0.10, 0.15 and 0.20 inches respectively. For each opening

that the valves held rigidly between the valve and its seat, air was blown through the valve at varying rates with varying resulting pressure drops up to 30 inches of water. The arrangement of apparatus for carrying out this part of the work is shown in Figs. 2 and 3. Fig. 2 is the arrangement used when the direction of

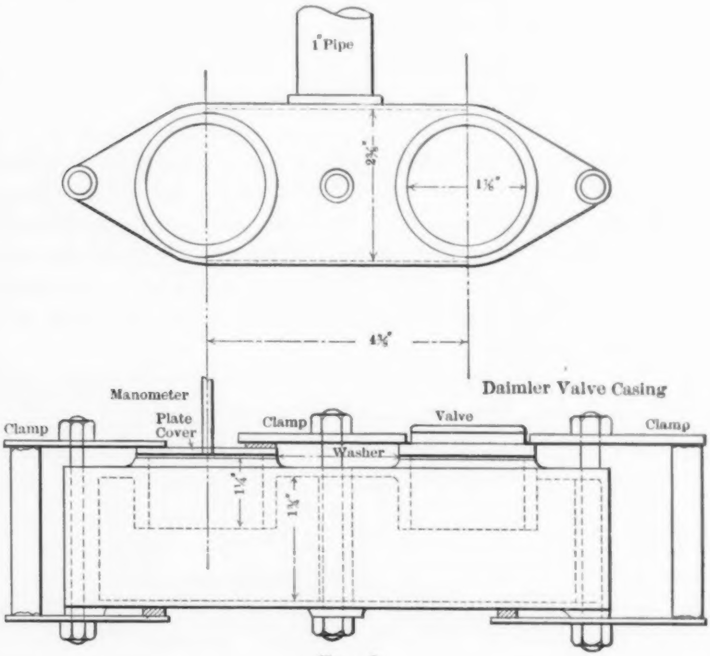


FIG. 2.

flow was from the inside to out, and Fig. 3 is the arrangement of both directions as the valve fastening here is reversible.

4. The conical seat valves were furnished by a Nash gas engine and had diameters of $1\frac{1}{2}$ and 2 inches. The small valve on the engine is used for the gas inlet and the larger for mixture inlet. The arrangement for steady flow runs are shown in Fig. 4.

5. Throughout this part of the work on steady flow of air, the air was measured before it reached the valve, and at a pressure higher than atmosphere, pressure drop occurring between meter line and atmosphere and the manometer indicating the pressure drop, as shown in Fig. 5, as used. Flow of air from inside out is equivalent to exhaust stroke. For determining the readings of

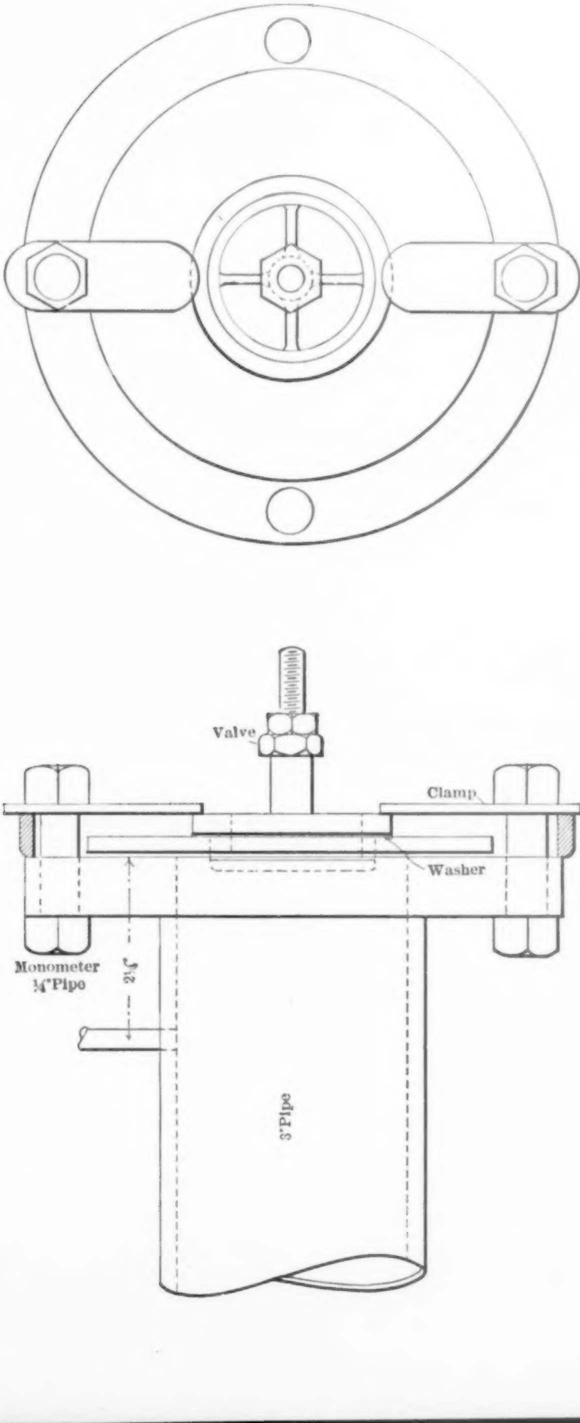


Fig. 3.

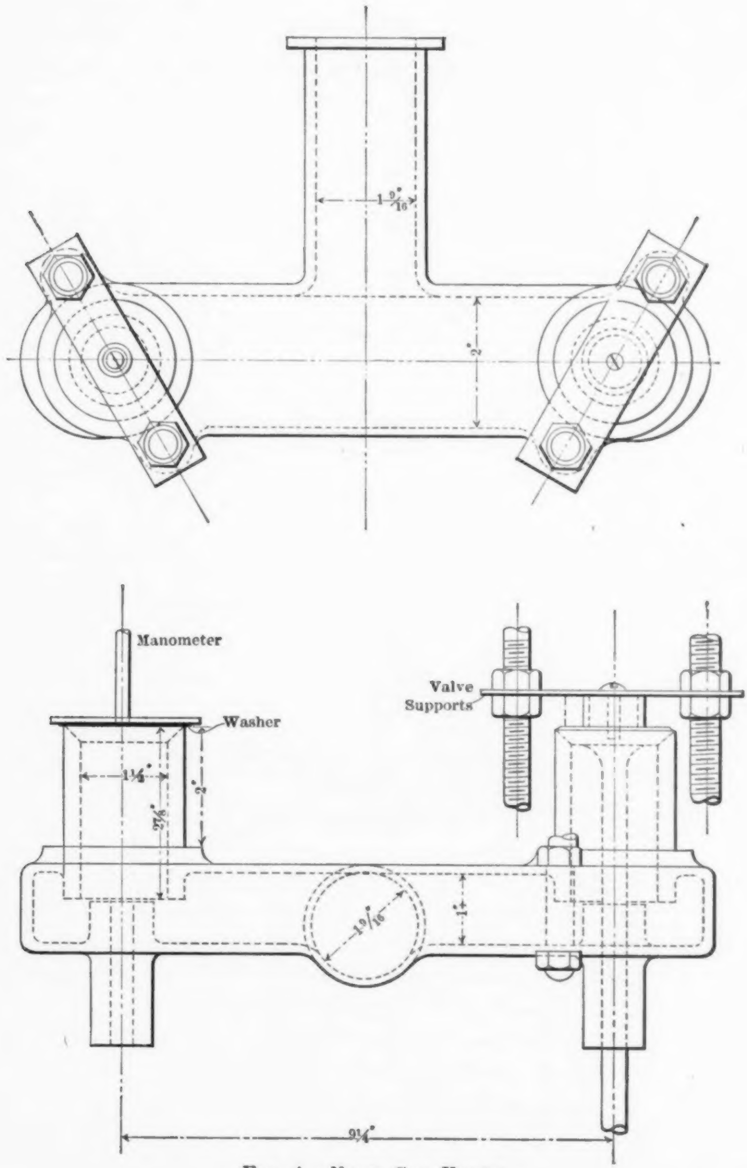


FIG. 4.—NASH GAS VALVE.

the meter and the pressure drop for the conical valve, when the flow of air was to be reversed; that is to say, in the exhaust valve direction the entire valve and piping, as shown in Fig. 4, was enclosed in a chamber, and air was supplied to this chamber. The

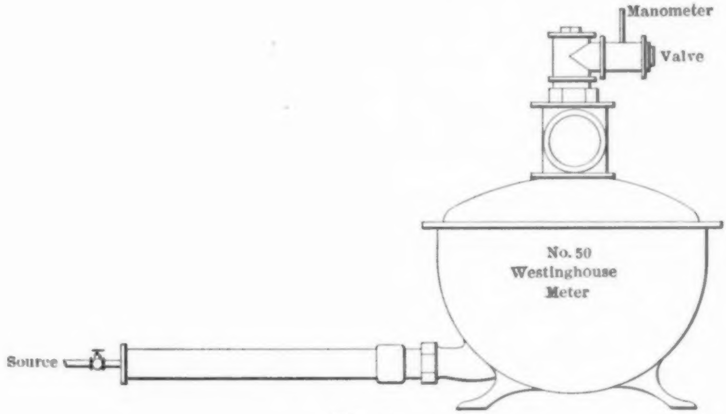


FIG. 5.

air then passed through the valve and out by a pipe which was used as a suction pipe on the engine.

6. The second series of runs was made with the valves located on the engines as they ordinarily operate. For this purpose a Daimler engine was piped to a meter so that it caused a slight back pressure. The meter therefore measured the discharge of air after exhaust.

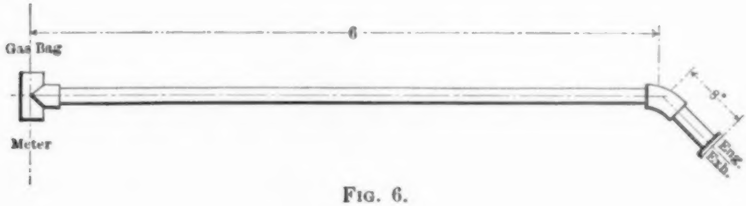


FIG. 6.

The engine was then driven by a belt from a steam engine, and runs were made covering a range of speeds. The valve being automatic it was provided with stops so that the greatest openings corresponded to those for the steady flow of runs, 0.05, 0.10, 0.15 and 0.20 inches. The actual lifts were drawn by an indicator, the piston rod being fastened to the valve stem. Pressure drops were

measured by indicators on the cylinder carrying a 10-pound spring. After the readings for this valve were obtained the meter position was changed so that the engine would draw air through the meter before it reached the cylinder. In this case the pressure drop was from atmosphere to something below, while in the other case it was from something above atmosphere to something less but still

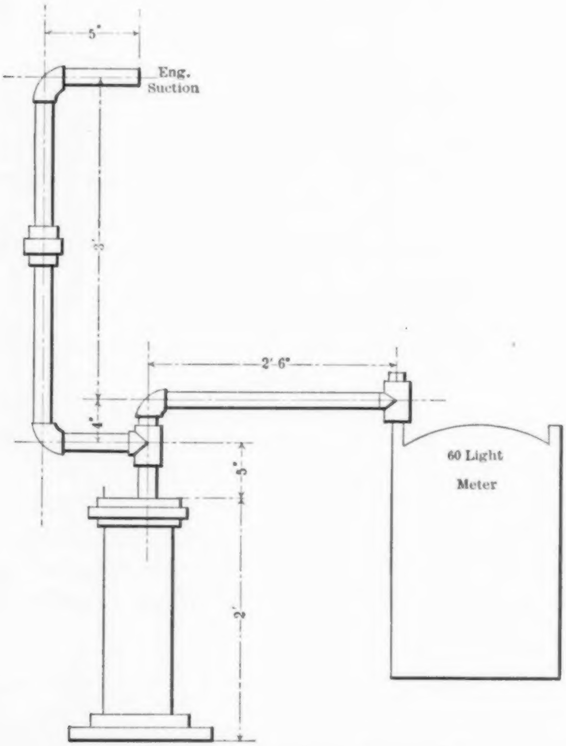


FIG. 7.—PIPING LAY-OUT FOR DAIMLER RUNS.

above atmosphere. As before, the engine was operated at a number of different speeds and the pressure drop through the exhaust valve was measured by the 10-pound spring indicator on the motor cylinder. This indicator card in fact gave the pressure drop through both valve courses on one diagram. The exhaust valve on this engine is mechanically opened by cams so that no stop could be used to adjust the openings to any desired limit. The actual lift of this valve was measured at any point of the stroke mechanically

by causing a pencil fastened to the stem to draw a line on the indicator drum so that the lift is given for every point of the stroke and having differing actual piston speeds.

7. Intermittent runs were also made on the Nash engine, on which all valves are mechanically operated. Here the valve runs were measured as on the exhaust valves of the Daimler by attaching a pencil to the valve stem. Pressure drops were shown as before, on the 10-pound scale indicator card from the motor cylinder. There were two inlet valve determinations. First for the small valve used on the engine for the gas inlet, and second for the larger valve controlling the mixture admission to the motor cylinder. These were not operated together. When the small one was being used the large one was held open and the air pipe plugged. When the large one was being used the small one was removed. In making the measurements on the exhaust valve the air for the engine, instead of being sent through the air pipe, was taken through the opening left by the removal of the small gas valve. The arrangement for this Nash engine run is shown in Figs. 8 and 9.

8. At each point of the stroke the piston speed can be found from the following table:

Suction Stroke Completed.	Piston Speed Factor.
.0000	0.000
.0796	0.587
.2878	0.954
.5505	1.000
.7878	0.778
.9455	0.413
1.0000	0.000

Exhaust Stroke Completed.	Piston Speed Factor.
.0000	0.000
.0543	0.413
.2122	0.778
.4495	1.000
.7122	0.954
.9204	0.587
1.0000	0.000

Actual piston speeds are to be found for any revolutions per minute by multiplying linear velocities of crank pin by the piston speed factors above. The Daimler engine had a bore of 3 15-16 inches and a stroke of 5 9-16 inches, giving a crank pin velocity of 145.6 feet per minute at 100 revolutions per minute.

9. The Nash engine had a bore of 6½ inches and a stroke of 10 inches, giving a crank pin velocity of 261.8 feet per minute at 100

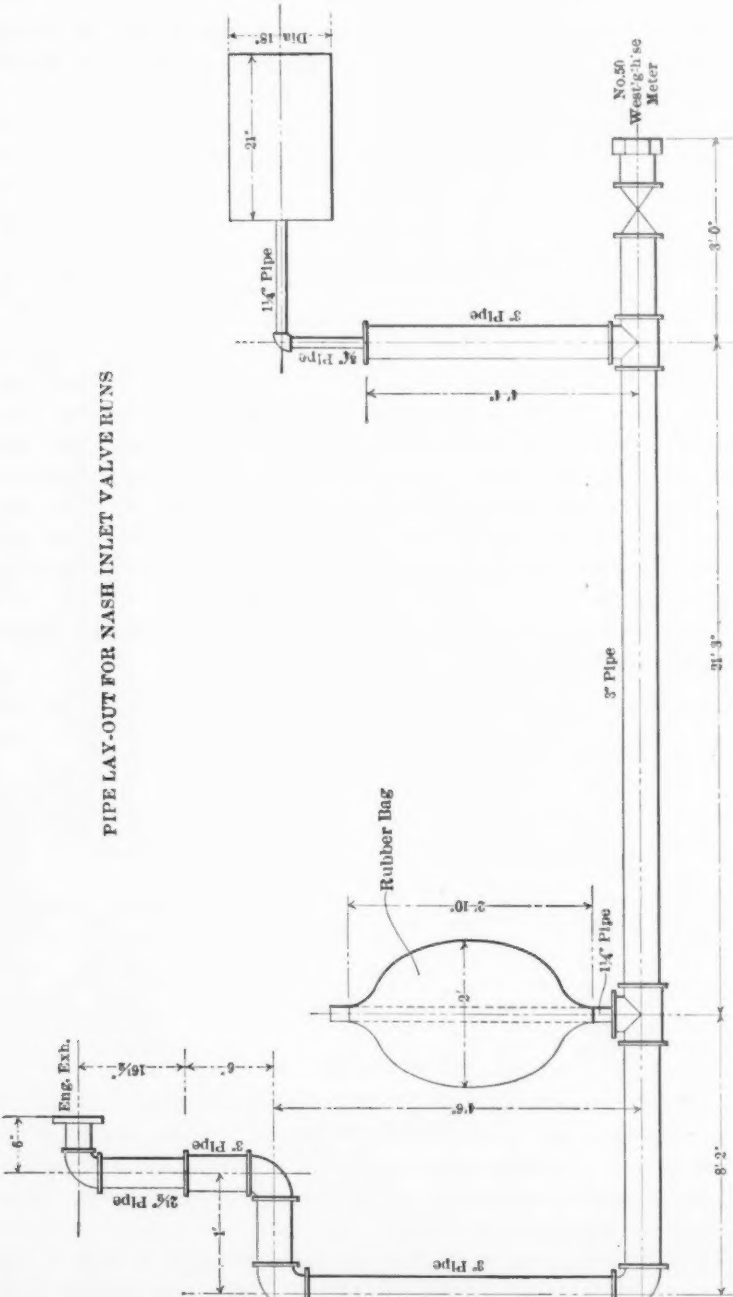

PIPE LAY-OUT FOR NASH INLET VALVE RUNS
Dia. 18"
21"
1¼" Pipe
¾" Pipe
3" Pipe
4'-4"
No.50
West'g'h'se
Meter
3'-0"
3" Pipe
21'-3"
Rubber Bag
2'-10"
2'
1¼" Pipe
Eng. Exh.
16½"
6"
4'6"
6"
3" Pipe
2½" Pipe
1'
3" Pipe
8'-2"

FIG. 8.

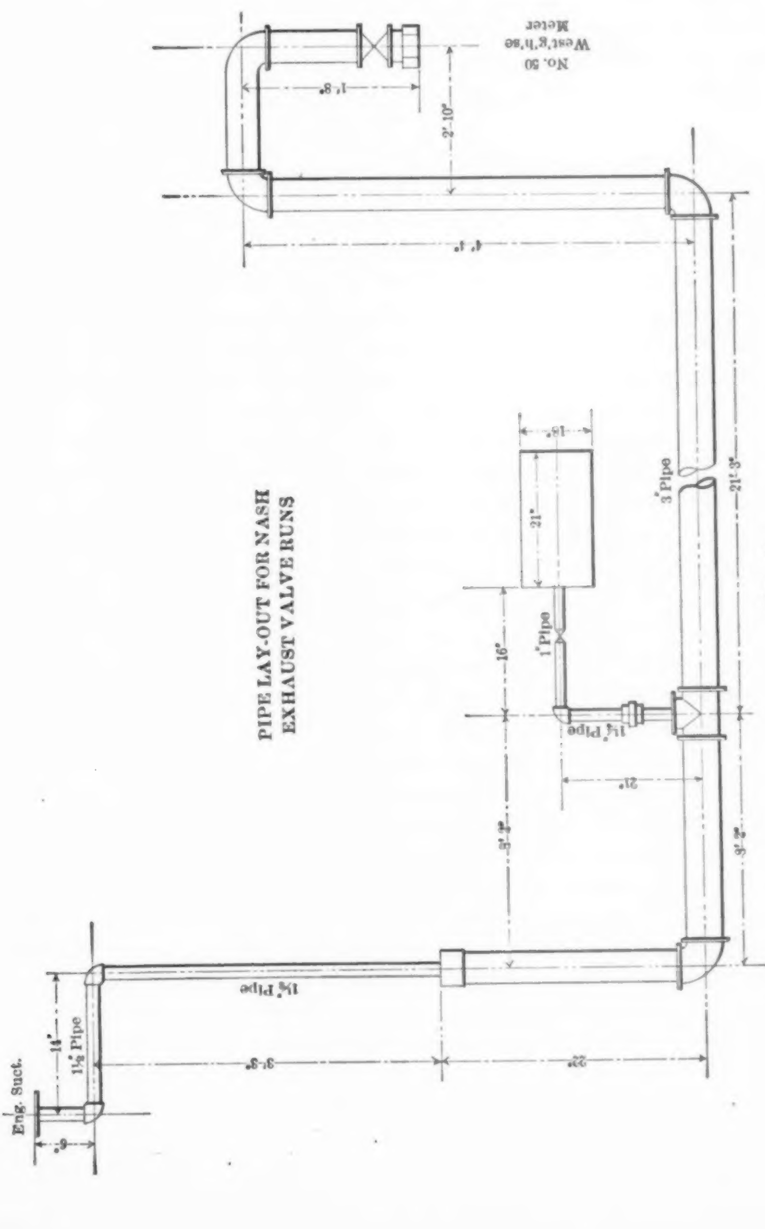

PIPE LAY-OUT FOR NASH
EXHAUST VALVE RUNS
No. 50
West'g'h'se
Meter
2' 10"
4' 1"
3" Pipe
21' 3"
21"
18"
16"
1" Pipe
1¼" Pipe
1½" Pipe
Eng. Suct.
14"
22"

FIG. 9.

revolutions per minute. Both engines had connecting rods five cranks long.

10. Valve openings are computed in the work for flat seats as the product of the circumference of the inner circle and valve lift, and for conical valves as by the formula:

$$A = \frac{\pi dh}{\sqrt{2}} + \frac{\pi h^2}{2\sqrt{2}} \text{ for small lifts.}$$

11. The first runs were made on the flat-seated Daimler valve, with a steady flow of air and the results are given in the curves and tables. In every case the density of the air measured by the meter is reduced to atmosphere and given both ways in the tables. The column called "Theoretical Velocity" is computed on air at atmospheric density by the usual formula of $V\sqrt{2gh}$. The ratio of the actual velocity to the theoretical velocity gives what is called the "Co-efficient of Efflux," and this appears for atmospheric air as well as for the air at meter density. It will be seen from the tables that the co-efficients computed in this way are nearly alike, and that the error in assuming the air measured by the meter to be not very different from the atmospheric air, is not great and the difference may in general be neglected. In the curves between the pressure drop and co-efficient of efflux it is seen that the co-efficient is greatest for the smallest opening and least for the largest opening. On the assumption of gas density being that of the higher pressure while passing through the valve, it appears from the co-efficient curves that the greater the pressure drop the smaller the co-efficient; on the contrary, when the density is that taken for atmospheric pressure, these co-efficients of efflux seem to be nearly constant for all pressure drops. The exhaust flow, or rather the flow in the opposite direction or flow in the exhaust direction, shows the results that seem to follow similar laws for the inlet, but the velocity and the co-efficients were smaller. The errors involved in the work are not greater than one per cent. for the meter observations, and for valve area computations on the flat valve may be neglected, but may be as much as ten per cent. for valve area computations on the conical valve, in spite of all care.

FLAT SEAT VALVE. (DIAMETER OF OPENING 1.58 inch.)

PART A-1. (CURVES FIG. 10.)

(Valve with housing as in Fig. 2.)

DIRECTION OF FLOW FROM INSIDE TO OUTSIDE OF VALVE, AS IN "INLET."

TABLE 1. RUN No. 1 (A-1).

Valve fixed at 0.05-inch lift. Area=0.001722 square feet.
Barometer=30.3 inches Hg. Average temperature of Air=74 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	5.13	2,980	.....	4,020	.740	.742
2	7.48	4,340	.....	5,685	.764	.768
3	9.34	5,420	.....	6,963	.779	.785
5	12.37	7,175	7,220	8,990	.804	.810
7	14.72	8,550	8,600	10,636	.808	.812
10	17.5	10,160	.....	12,713	.799	.819
15	21.1	12,250	12,300	15,570	.790	.820
20	24.0	13,930	.....	17,978	.775	.815
25	26.4	15,330	.....	20,100	.763	.812
30	28.5	16,550	.....	22,019	.750	.808

TABLE 2. RUN No. 2 (A-1).

Valve fixed at 0.10-inch lift. Area=0.003444 square feet.
Barometer=30 inches Hg. Average temperature of Air = 82 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	9.3	2,700	.....	4,020	.671	.673
2	13.31	3,863	.....	5,685	.680	.683
3	16.4	4,760	.....	6,963	.685	.690
5	21.1	6,125	.....	8,990	.681	.689
7	24.79	7,180	7,220	10,636	.678	.690
10	29.5	8,550	.....	12,713	.672	.686
15	35.6	10,250	10,320	15,570	.664	.689
20	40.0	11,600	11,700	17,978	.651	.685
25	44.4	12,880	.....	20,100	.640	.681
30	47.6	13,810	.....	22,019	.626	.674

TABLE 3. RUN No. 3 (A-1).

Valve fixed at 0.15-inch lift. Area of opening = 0.005166 square feet.
Barometer = 29.77. Average temperature of Air = 77 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	13.47	2,600	.....	4,020	.647	.649
2	18.93	3,660	.....	5,685	.644	.647
3	23.00	4,445	.....	6,963	.638	.643
5	29.50	5,710	.....	8,990	.635	.643
7	34.70	6,715	.....	10,636	.631	.642
10	41.10	7,950	.....	12,713	.624	.640
15	50.00	9,670	9,520	15,570	.612	.634

TABLE 4. RUN No. 4 (A-1).

Valve fixed at 0.20-inch lift. Area of opening = 0.006888 square feet.
Barometer = ———. Average temperature of Air = 75 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	16.9	2,455	.....	4,020	.610	.612
2	23.62	3,435	.....	5,685	.603	.606
3	29.28	4,250	.....	6,963	.610	.615
4	33.3	4,840	.....	8,040	.602	.608
6	40.28	5,850	.....	9,847	.595	.604
8	46.00	6,680	.....	11,370	.588	.600

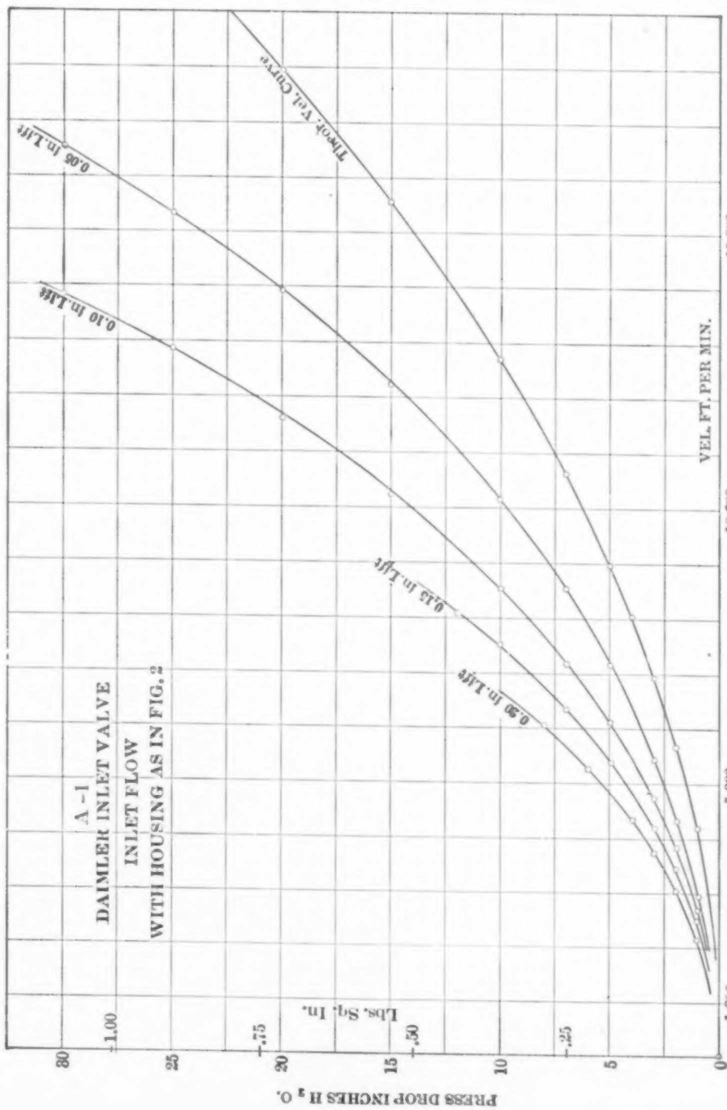

A-1
DAIMLER INLET VALVE
INLET FLOW
WITH HOUSING AS IN FIG. 2
Theor. Vel. Curve
0.05 In. Lift
0.10 In. Lift
0.15 In. Lift
0.20 In. Lift
VEL. FT. PER MIN.
1,000
5,000
10,000
15,000
Lbs. Sq. In.
1.00
.75
.50
.25
PRESS DROP INCHES H_2O.
30
25
20
15
10
5
0

Fig. 10.

Part A-2. (Curves Fig. 11.)

(Valve without housing as in Fig. 3.)

Direction of Flow from Inside to Outside of Valve as in "Inlet."

Table 5. Run No. 1 (A-2).

Valve fixed at 0.05-inch lift. Area of opening = 0.001722 square feet.
Barometer 29.65 inches Hg. Average temperature of Air = 73 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	5.48	3,180	.....	4,020	.790	.792
2	7.79	4,520	.....	5,685	.796	.800
3	9.41	5,460	5,600	6,963	.805	.811
5	12.53	7,275	.....	8,989	.808	.819
7	14.79	8,590	.....	10,636	.808	.823
10	17.6	10,210	.....	12,713	.803	.824
15	21.25	12,340	.....	15,570	.794	.825
20	23.25	13,540	14,100	17,978	.784	.824
25	25.98	15,070	15,550	20,100	.774	.823
30	28.18	16,340	16,810	22,019	.764	.822

Table 6. Run No. 2 (A-2).

Valve fixed at 0 10-inch lift. Area of opening = 0.003444 square feet.
Barometer = 29.81 inches Hg. Average temperature of Air = 76 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	9.49	2,750	.....	4,020	.684	.686
2	13.18	3,820	.....	6,100	.672	.676
3	16.05	4,660	4,710	.....	.677	.682
5	21.72	6,290	6,100	8,990	.679	.683
7	24.3	7,050	7,150	10,636	.672	.684
10	28.83	8,370	8,480	12,713	.667	.684
15	35.40	10,280	10,220	15,570	.658	.683
20	40.00	11,600	11,650	17,778	.648	.681
25	44.40	12,880	.....	20,100	.640	.681

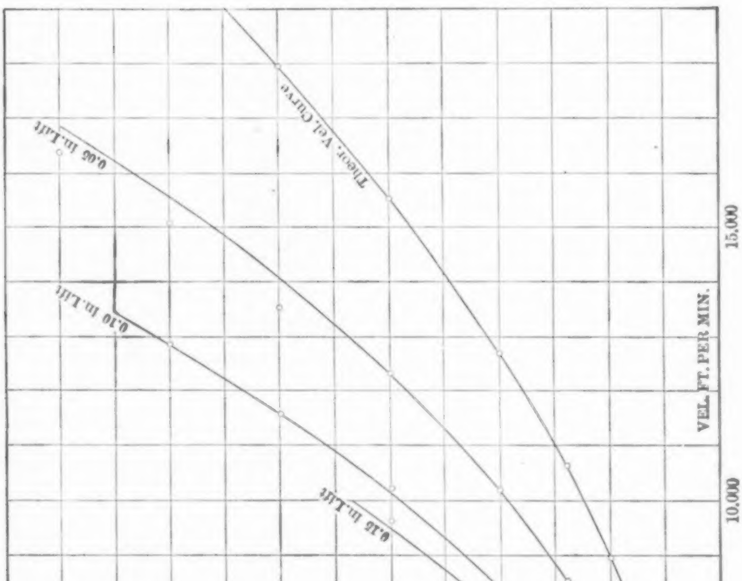

A-2
DAIMLER INLET VALVE
INLET FLOW
AS IN FIG. 3
0.05 In. Lift
0.10 In. Lift
0.15 In. Lift
0.20 In. Lift
Theor. Vel. Curve
VEL. FT. PER MIN.
1,000
5,000
10,000
15,000
PRESS DROP INCHES H2O.
30
25
20
15
10
5
Lbs. Sq. In.
1.00
.75
.50
.25

FIG. 11.

TABLE 7. RUN No. 3 (A-2).

Valve fixed at 0.15-inch lift. Area = 0.005166 square feet.
Barometer 30.3 inches Hg. Average temperature of Air = 77 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	12.92	2,500	2,450	4,020	.609	.611
2	17.9	3,461	.....	5,685	.609	.612
3	21.70	4,200	.....	6,963	.603	.608
5	27.95	5,410	.....	8,990	.602	.609
7	32.95	6,375	.....	10,636	.600	.610
10	38.9	7,525	.....	12,713	.592	.607
12	41.85	8,099	8,150	13,926	.585	.603

TABLE 8. RUN No. 4 (A-2).

Valve fixed at 0.20-inch lift. Area of opening = 0.006888 square feet.
Barometer 30.25 inches Hg. Average temperature of Air = 75 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	14.88	2,160	.....	4,020	.537	.538
2	21.05	3,060	.....	5,685	.539	.542
3	25.8	3,750	.....	6,963	.539	.543
4	29.95	4,350	.....	8,040	.541	.546
6	36.3	5,270	.....	9,847	.536	.544
8	41.8	6,070	.....	11,370	.534	.545

PART B. (CURVES FIG. 12.)

(Valve without housing as in Fig. 3.)

DIRECTION OF FLOW FROM OUTSIDE TO INSIDE OF VALVE AS IN EXHAUST.

TABLE 9. RUN No. 1 (B).

Valve fixed at 0.05-inch lift. Area of opening = 0.001722 square feet.
Barometer 30 inches Hg. Average temperature of Air = 80 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	5.29	3,070	2,900	4,020	.72	.722
2	7.39	4,290	.....	5,685	.756	.760
3	9.29	5,390	.....	6,963	.773	.778
5	12.00	6,970	7,000	8,990	.778	.788
7	13.85	8,004	.....	10,636	.752	.769
10	16.94	9,840	.....	12,713	.773	.793
15	20.53	11,900	.....	15,570	.765	.794
20	23.30	13,520	.....	17,918	.753	.792
25	25.58	14,810	14,920	20,100	.743	.791
30	27.80	16,120	.....	22,019	.732	.808

TABLE 10. RUN No. 2 (B).

Valve fixed at 0.10-inch lift. Area of opening = 0.003444 square feet.
Barometer 29.89 inches Hg. Average temperature of Air = 81 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	8.08	2,345	.....	4,020	.584	.585
2	11.54	3,350	.....	5,685	.590	.593
3	14.03	4,080	.....	6,963	.586	.590
5	18.23	5,300	.....	8,990	.590	.598
7	21.50	6,240	.....	10,636	.586	.597
10	25.70	7,460	.....	12,713	.586	.601
15	31.20	9,060	.....	15,570	.582	.604
20	35.70	10,380	10,300	17,978	.573	.602
25	39.30	11,400	.....	20,100	.568	.604
30	42.70	12,400	.....	22,019	.563	.607

TABLE 11. RUN No. 3 (B).

Valve fixed at 0.15-inch lift. Area of opening = 0.005166.
Barometer = 30.20 inches Hg. Average temperature of Air = 75 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	11.31	2,190	.....	4,020	.544	.545
2	15.88	3,070	.....	5,685	.540	.543
3	19.38	3,750	.....	6,963	.539	.543
5	24.70	4,780	4,800	8,990	.534	.541
7	29.28	5,670	.....	10,636	.533	.543
10	35.10	6,800	6,750	12,713	.532	.546
15	41.20	7,970	8,120	15,570	.522	.542
20	47.65	7,220	.....	17,978	.514	.540

TABLE 12. RUN No. 4 (B).

Valve fixed at 0.20-inch opening. Area of opening = 0.006888 square feet.
Barometer = 30.20 inches Hg. Average temperature of Air = 74 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	14.07	2,022	.....	4,020	.503	.504
2	19.48	2,822	.....	5,685	.497	.499
3	23.75	3,450	.....	6,963	.495	.499
4	27.3	3,965	.....	8,040	.494	.499
6	33.35	4,840	.....	9,847	.492	.499
8	38.28	5,560	.....	11,370	.490	.500
10	42.75	6,210	.....	12,713	.488	.500
12	46.00	6,680	.....	13,926	.479	.494
15	51.00	7,410	.....	15,570	.477	.495

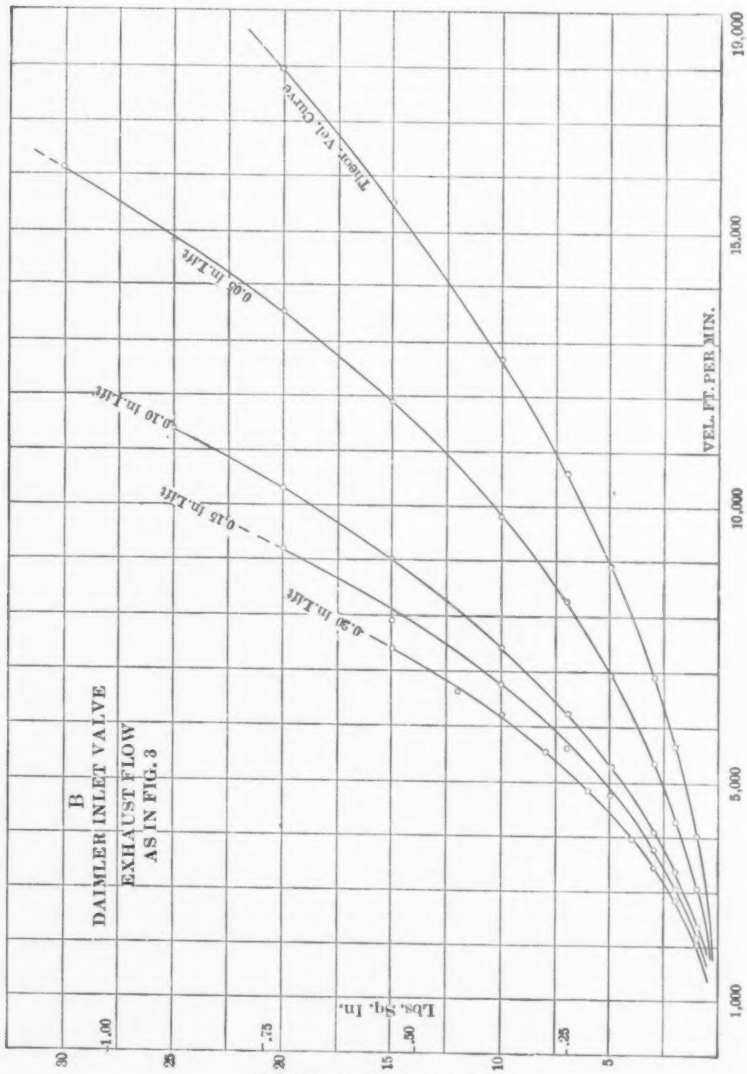

B
DAIMLER INLET VALVE
EXHAUST FLOW
AS IN FIG. 3
Theor. Vel. Curve
0.05 In. Lift
0.10 In. Lift
0.15 In. Lift
0.20 In. Lift
VEL. FT. PER MIN.
1,000
5,000
10,000
15,000
19,000
PRESS DROP INCHES H_2O.
30
25
20
15
10
5
Lbs. Sq. In.
1.00
.75
.50
.25

Fig. 12.

12. The second series of runs was made on steady flow from conical valves and velocities were computed as explained.

CONICAL SEAT VALVES.

(Diameter of opening = 1.5 inches.)

DIRECTION OF FLOW FROM INSIDE TO OUTSIDE OF VALVE AS IN "INLET."

TABLE 13. RUN 1 (A) (CURVES FIG. 13.)

Valve fixed at 0.05-inch lift. Area of opening = 0.001351 square feet.
Barometer 30.00 inches Hg. Average temperature of Air = 77 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	4.21	3,160	.....	4,020	.786	.788
2	6.30	4,660	4,460	5,685	.820	.824
4	9.41	6,960	.....	8,040	.866	.875
6	11.75	8,690	.....	9,847	.884	.897
9	14.50	10,720	.....	12,060	.890	.910
12	16.80	12,430	.....	13,926	.893	.919
15	18.75	13,880	.....	15,570	.890	.923
20	21.60	15,990	.....	17,978	.889	.935
25	23.95	17,710	.....	20,100	.882	.938
30	26.00	19,230	.....	22,019	.873	.940
2½ inches Hg.	27.65	20,250	.....	23,420	.864	....
3 inches Hg.	30.00	22,200	.....	25,370	.875	....
3½ inches Hg.	31.80	23,530	.....	27,720	.849	....
4 inches Hg.	33.5	24,800	.....	29,630	.836	....
5 inches Hg.	36.8	27,210	.....	33,130	.823	....
6 inches Hg.	38.7	28,620	.....	36,302	.789	....

TABLE 14. RUN NO. 2 (A).

Valve fixed at 0.10-inch lift. Area of opening = 0.002391 square feet.
Barometer = 29.93 inches Hg. Average temperature of Air = 80 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	9.19	3,841	.....	4,020	.925	.927
2	13.20	5,520	.....	5,685	.972	.977
3	16.44	6,875	.....	6,963	.984	.991
5	21.40	8,950	8,850	8,990	.985	.998
9	29.00	12 120	12,000	12,060	.994	1.013
14	35.68	15,400	14,800	15,042	.984	1.019
20	42.20	17,650	.....	17,978	.982	1.033
25	46.50	19,450	.....	20,100	.968	1.030
30	50.20	21,000	.....	22,019	.954	1.027

TABLE 15. RUN NO. 3 (A).

Valve fixed at 0.15-inch lift. Area of opening = 0.00364 square feet.
Barometer = 30.17 inches Hg. Average temperature of Air = 76 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	13.68	3,755	.....	4,020	.935	.931
2	19.45	5,340	.....	5,685	.940	.945
3	23.80	6,540	.....	6,963	.940	.947
4	27.40	7,525	7,600	8,084	.940	.949
6	33.85	9,300	.....	9,847	.945	.959
8	38.85	10,680	.....	11,370	.940	.959
10	42.40	11,650	11,850	12,713	.932	.956
13	48.60	13,350	13,400	14,494	.926	.957
15	52.00	14,280	14,300	15,570	.919	.954

TABLE 16. RUN NO. 4 (A).

Valve fixed at 0.20-inch lift. Area of opening = 0.00494 square feet.
Barometer = 30.17 inches Hg. Average temperature of Air = 78 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	17.3	3,500	.....	4,020	.870	.872
2	24.25	4,910	.....	5,685	.864	.868
3	29.50	5,970	.....	6,963	.857	.864
4	33.85	6,850	.....	8,040	.853	.861
6	41.50	8,400	.....	9,844	.854	.867
8	47.05	9,520	.....	11,370	.838	.855
10	52.60	10,650	.....	12,713	.838	.859

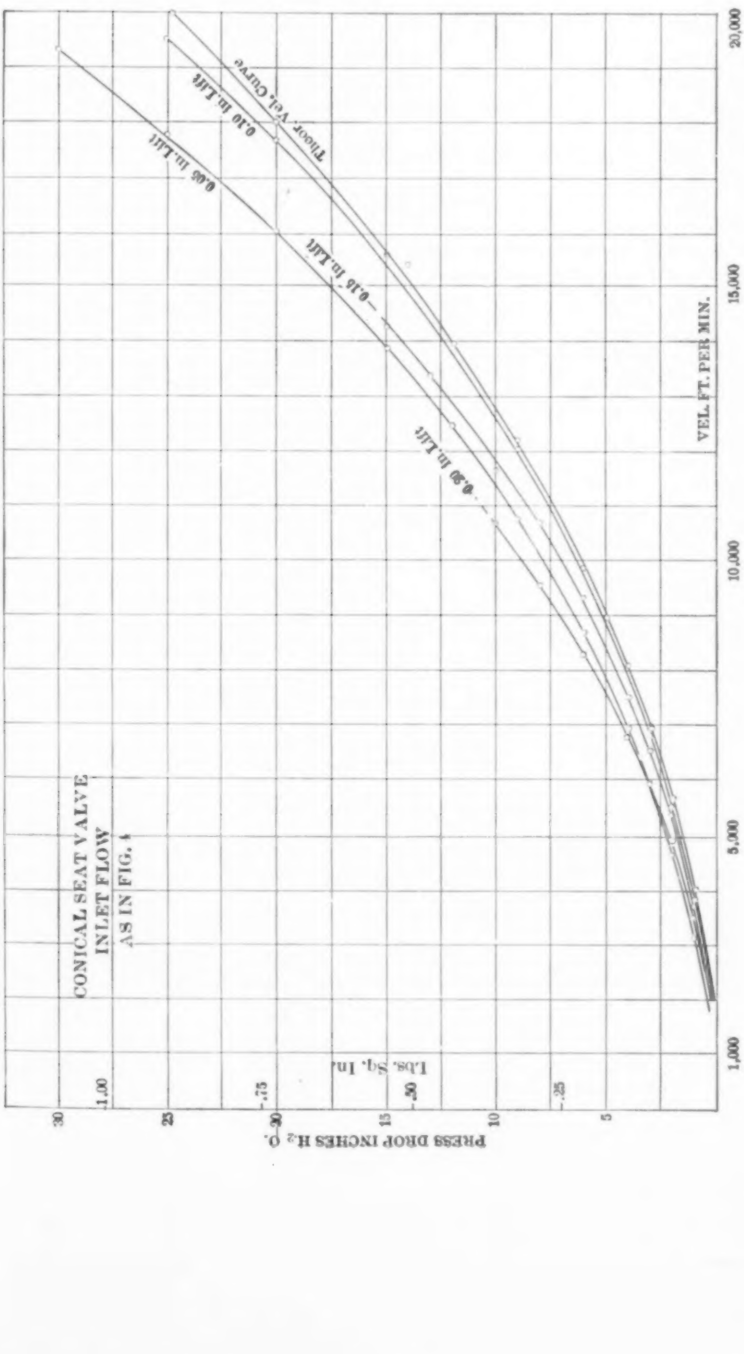

CONICAL SEAT VALVE
INLET FLOW
AS IN FIG. 4
Theor. Vel. Curve
0.10 in. Lift
0.05 in. Lift
0.15 in. Lift
0.20 in. Lift
VEL. FT. PER MIN.
1,000
5,000
10,000
15,000
20,000
PRESS DROP INCHES H_2O.
30
25
20
15
10
5
Lbs. Sq. In.
1.00
.75
.50
.25

Fig. 13.

CONICAL SEAT VALVE.

Part B.

(Valve with housing as in Fig. 4.)

Direction of Flow from Outside to Inside of Valve as in "Exhaust."

Table 17. Run No. 1 (B). (Curves Fig. 15.)

Valve fixed at 0.05-inch lift. Area of opening = 0.001351 square feet.
Barometer = 29.69 inches Hg. Average temperature of Air = 77 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	3.88	2,870	2,900	4,020	.721	.723
2	5.77	4,270	.....	5,685	.752	.756
3	7.15	5,280	.....	6,963	.758	.764
5	9.40	6,960	.....	8,990	.784	.794
7	11.22	8,300	8,200	10,636	.771	.785
10	13.48	9,960	9,880	12,713	.776	.796
12	14.75	10,910	10,800	13,926	.775	.799
15	16.40	12,120	.....	15,570	.779	.809
20	18.94	14,100	13,950	17,978	.777	.817
25	20.92	15,470	15,580	20,100	.775	.825
30	22.85	16,910	.....	22,019	.767	.827

Table 18. Run No. 2 (B).

Valve fixed at 0.10-inch lift. Area of opening = 0.002391 square feet.
Barometer = 29.81 inches Hg. Average temperature of Air = 77 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Coefficient of Efflux. (Air at Meter Density.)	Coefficient of Efflux. (Air at Atmospheric Density.)
1	8.56	3,580	.....	4,020	.891	.893
2	12.18	5,080	.....	5,685	.894	.898
3	14.92	6,250	.....	6,963	.898	.907
5	19.49	8,150	.....	8,990	.906	.918
7	22.82	9,560	.....	10,636	.899	.915
10	27.21	11,380	.....	12,715	.894	.917
15	33.18	13,880	.....	15,570	.891	.925
20	37.68	15,780	.....	17,978	.878	.923
25	41.30	17,290	17,400	20,100	.865	.920
30	44.30	18,550	18,800	22,019	.855	.922

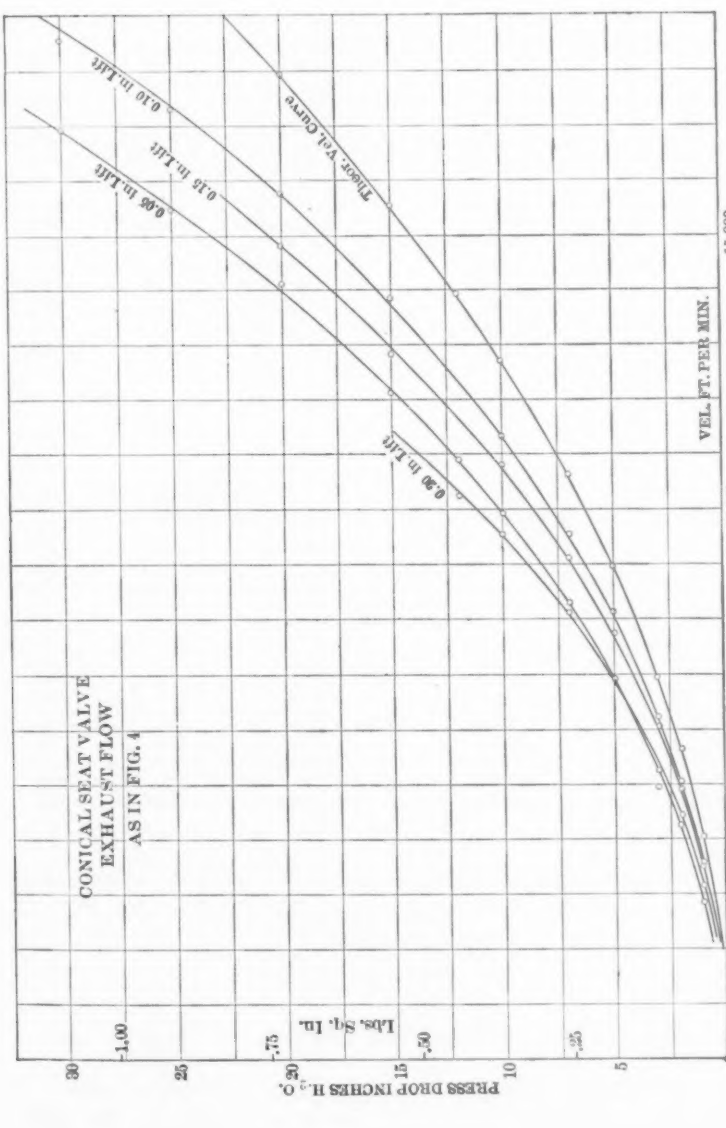

CONICAL SEAT VALVE
EXHAUST FLOW
AS IN FIG. 4
Theor. Vel. Curve
0.10 In. Lift
0.15 In. Lift
0.05 In. Lift
0.20 In. Lift
VEL. FT. PER MIN.
1,000
5,000
10,000
15,000
PRESS DROP INCHES H_2O.
30
25
20
15
10
5
Lbs. Sq. In.
1.00
.75
.50
.25

Fig. 15.

TABLE 19. RUN No. 3 (B).

Valve fixed at 0.15-inch lift. Area of opening = 0.00364 square feet. Barometer = 29.84 inches Hg. Average temperature of Air =.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	12.83	3,525	.....	4,020	.878	.880
2	17.90	4,920	.....	5,685	.866	.870
3	22.00	6,040	.....	6,963	.867	.873
5	28.39	7,780	.....	8,990	.867	.878
7	33.32	9,120	.....	10,636	.857	.872
10	39.38	10,800	.....	12,713	.849	.871
15	46.70	12,820	13,000	15,570	.835	.867
20	53.85	14,800	.....	17,978	.824	.866

TABLE 20. RUN No. 4 (B).

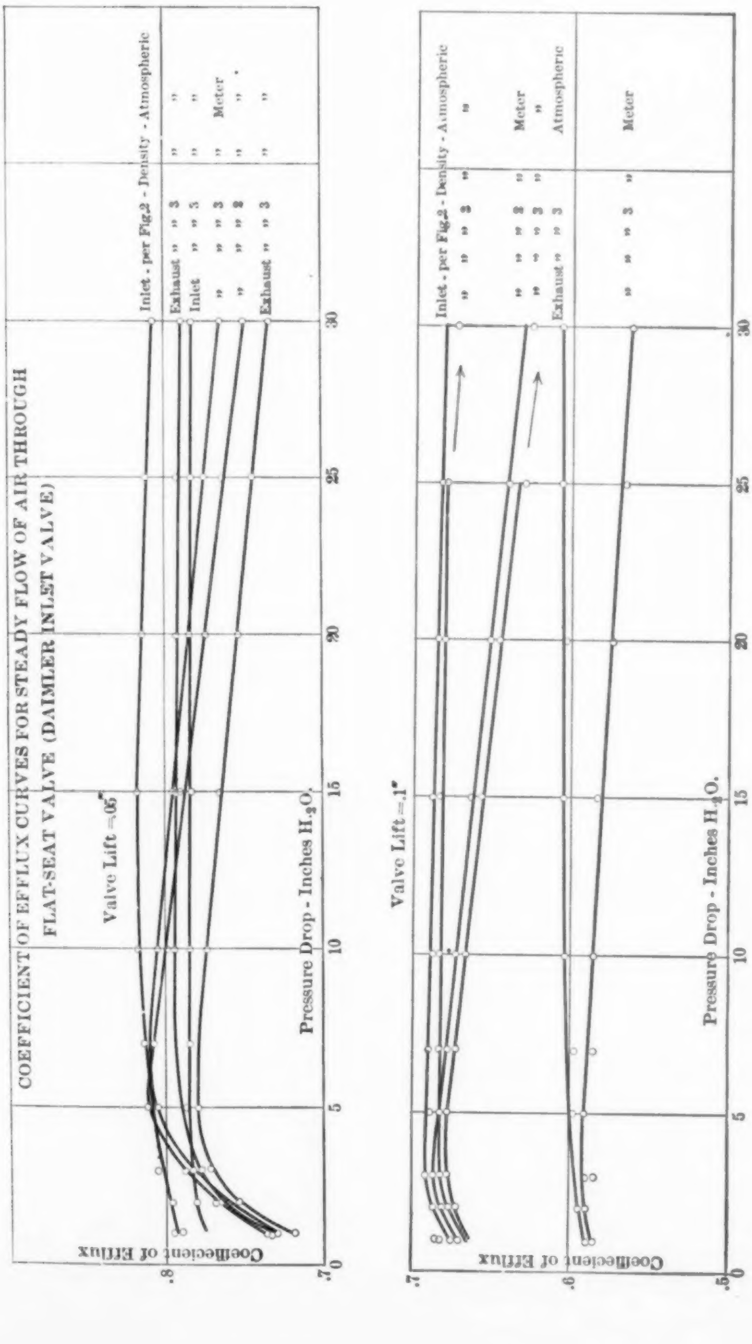
Valve set at 0.20-inch lift. Area of opening = 0.00494 square feet. Barometer ————. Average temperature of Air = 82 degrees Fahr.

Pressure Drop. (Inches of Water.)	Cubic Feet of Air per Minute.	Velocity. (Ft. per Min.)	Velocity corrected from Curves. (Ft. per Min.)	Theoretical Velocity at 72° Fahr. (Ft. per Min.)	Co-efficient of Efflux. (Air at Meter Density.)	Co-efficient of Efflux. (Air at Atmospheric Density.)
1	15.51	3,140	.....	4,020	.780	.782
2	21.9	4,430	.....	5,686	.780	.784
3	24.4	4,940	5,300	6,963	.760	.766
5	34.40	6,960	.....	8,990	.775	.785
7	40.25	8,150	.....	10,636	.766	.780
10	47.25	9,560	.....	12,713	.751	.770
12	50.85	10,290	.....	13,926	.738	.761

13. To most effectively obtain the co-efficient of efflux and their variations with conditions they are plotted in two sets, Fig. 16 for flat valves and Fig. 17 for conical.

INTERMITTENT FLOW.

14. As actually used in the engine the detrimental effect of suction pressure drop is first lost work measured by the mean suction pressure taken by the planometer from the pressure drop diagram, and second a lost volume or decreased volumetric cylinder efficiency shown by the fraction of stroke at which the com-


COEFFICIENT OF EFFLUX CURVES FOR STEADY FLOW OF AIR THROUGH
FLAT-SEAT VALVE (DAIMLER INLET VALVE)
Valve Lift =.05"
Coefficient of Efflux
Pressure Drop - Inches H_2O.
Inlet - per Fig.2 - Density - Atmospheric
Exhaust " " 3 " "
Inlet " " 3 " "
" " " 3 " Meter
" " " 3 " "
Exhaust " " 3 " "
Valve Lift =.1"
Inlet - per Fig.2 - Density - Atmospheric
" " " 3 " "
" " " 3 " Meter
" " " 3 " "
Exhaust " " 3 " Atmospheric
" " " 3 " Meter
.8
.7
.6
.5
0
5
10
15
20
25
30

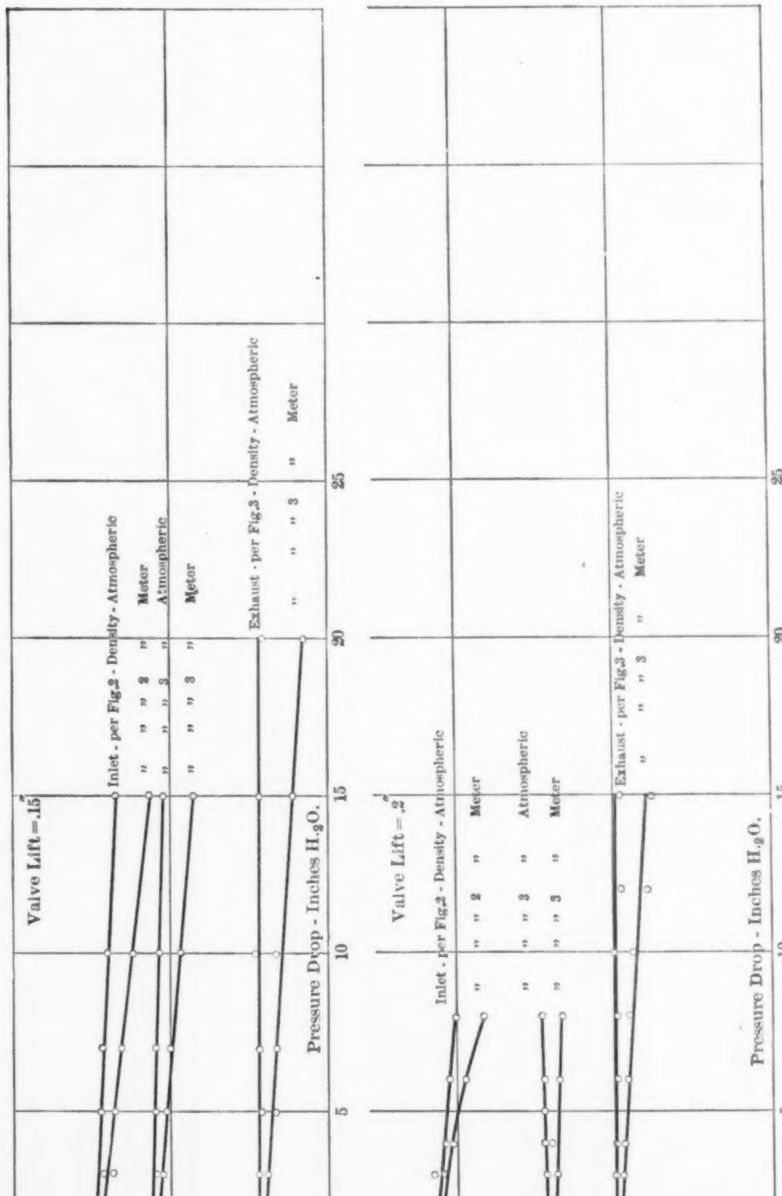
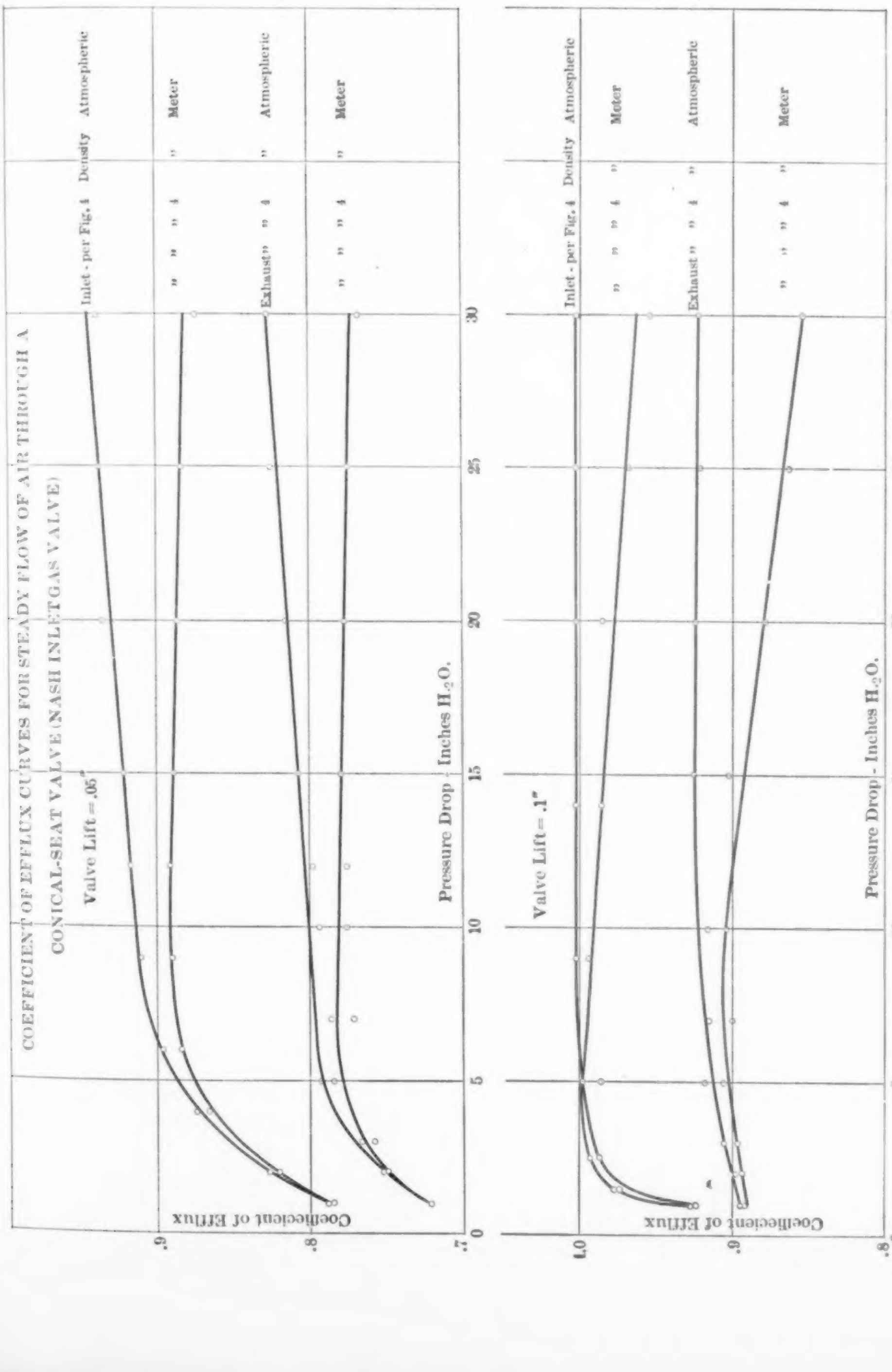

Valve Lift = .15"
Inlet - per Fig.2 - Density - Atmospheric
" " " 2 " Meter
" " " 3 " Atmospheric
" " " 3 " Meter
Exhaust - per Fig.3 - Density - Atmospheric
" " " 3 " Meter
Coefficient of Efflux
Pressure Drop - Inches H_2O.
Valve Lift = .2"
Inlet - per Fig.2 - Density - Atmospheric
" " " 2 " Meter
" " " 3 " Atmospheric
" " " 3 " Meter
Exhaust - per Fig.3 - Density - Atmospheric
" " " 3 " Meter
Coefficient of Efflux
Pressure Drop - Inches H_2O.

Fig. 16.


COEFFICIENT OF EFFLUX CURVES FOR STEADY FLOW OF AIR THROUGH A
CONICAL-SEAT VALVE (NASH INLET GAS VALVE)
Valve Lift = .05"
Inlet - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Exhaust " " 4 " Atmospheric
" " " 4 " Meter
Coefficient of Efflux
Pressure Drop - Inches H_2O.
Valve Lift = .1"
Inlet - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Exhaust " " 4 " Atmospheric
" " " 4 " Meter
Coefficient of Efflux
Pressure Drop - Inches H_2O.

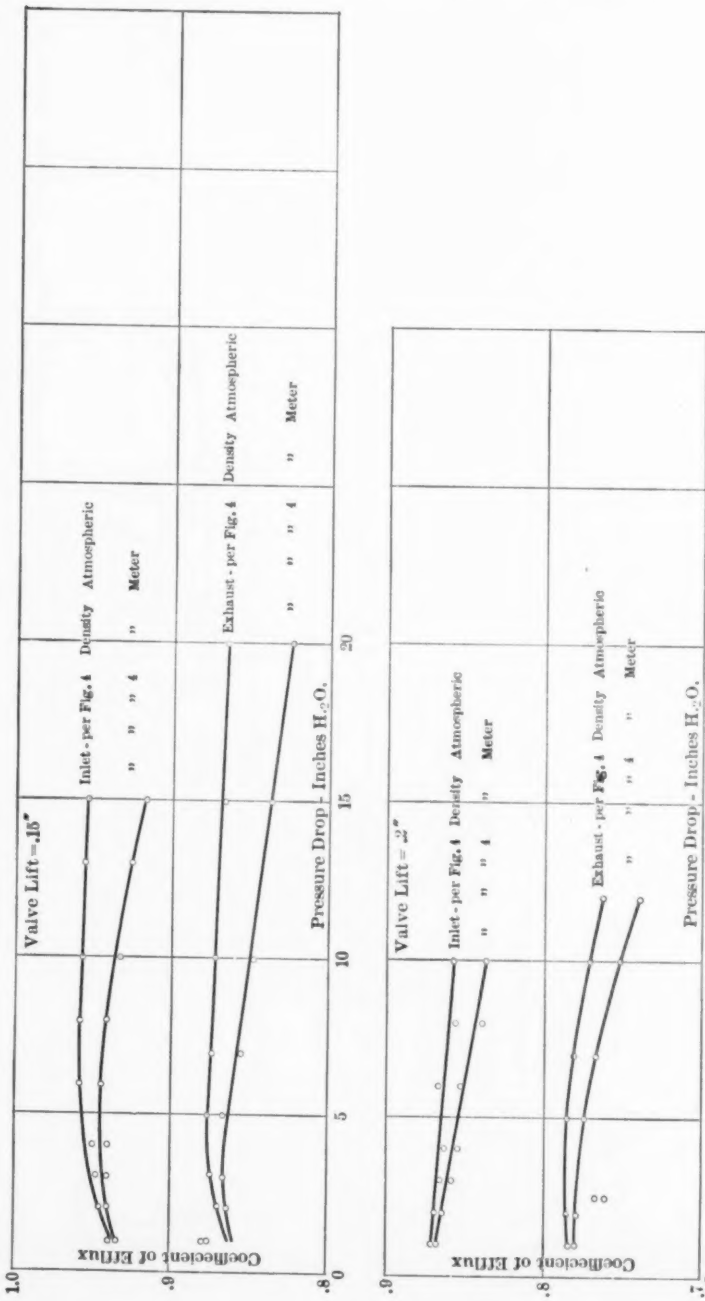

Valve Lift = .15"
Inlet - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Exhaust - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Coefficient of Efflux
Pressure Drop - Inches H2O.
1.0
.9
.8
0
5
10
15
20
Valve Lift = .2"
Inlet - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Exhaust - per Fig. 4 Density Atmospheric
" " " 4 " Meter
Coefficient of Efflux
Pressure Drop - Inches H2O.
.9
.8
.7
0
5
10
15

FIG. 17.

pression line crosses atmosphere. The values for these are given in the tables.

Similarly for exhaust or discharge the first effect is one of resistance measured by the mean pressure above atmosphere on the pressure drop diagram. In gas engines, however, the terminal exhaust pressure measures the dilution of fresh charge by hot burnt gases. This is also given from the indicator cards.

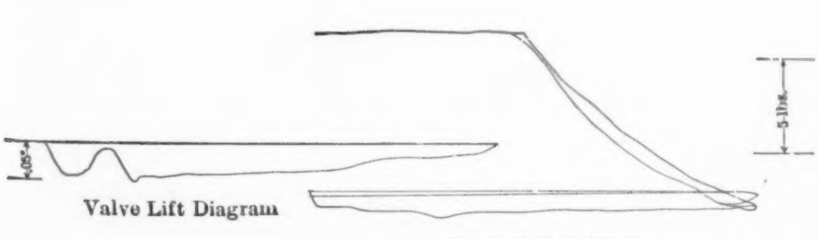
INTERMITTENT FLOW. (FLAT SEAT INLET VALVE. DAIMLER ENGINE.)

TABLE 21. RUN NO. 1 (A).

Maximum inlet valve lift 0.05. Average temperature of air = 77 degrees Fahr.

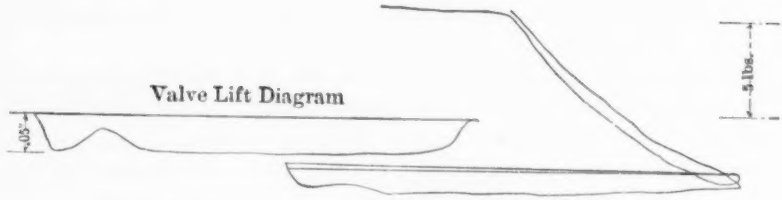
R. P. M.	Cubic Feet of Air pumped per Min.	Piston Displacement. (Cu. Ft. per Min.)	Per Cent. lost Volume.	Mean Resistance to Suction.
150	2.72	2.94	7.50	.71
200	3.53	3.92	9.95	1.06
250	4.23	4.90	13.68	1.22
300	5.40	5.88	8.17	1.54
350	6.21	6.86	9.48	2.
400	7.10	7.84	9.44	2.42
450	8.06	8.82	8.62	2.75
500	8.95	9.80	8.68	3.08
575	9.90	11.25	12.00	3.4

Valve lift and pressure drop diagrams, Figs. 18 to 26.



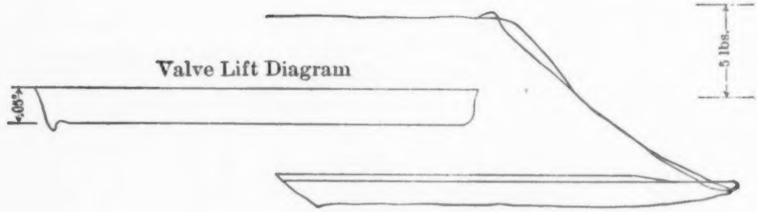
Pressure Drop Diagram
0.05" Max. Lift
150 R.P.M.

FIG. 18.



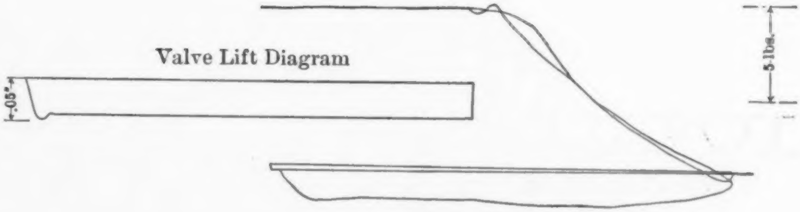
Pressure Drop Diagram
0.05-inch Max. Lift
200 R.P.M.

Fig. 19.



Pressure Drop Diagram
0.05-inch Max. Lift
250 R.P.M.

Fig. 20.



Pressure Drop Diagram
0.05-inch Max. Lift
300 R.P.M.

Fig. 21.

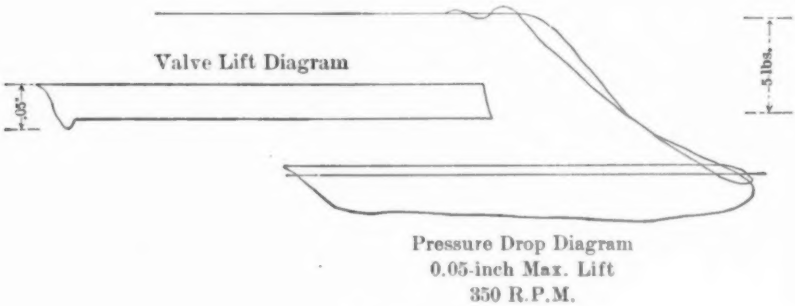


FIG. 22.

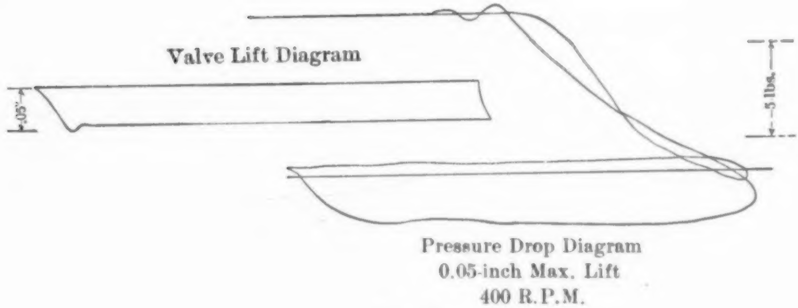


FIG. 23.

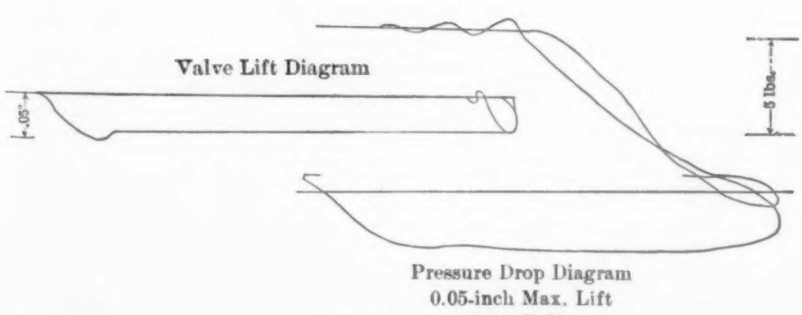


FIG. 24.

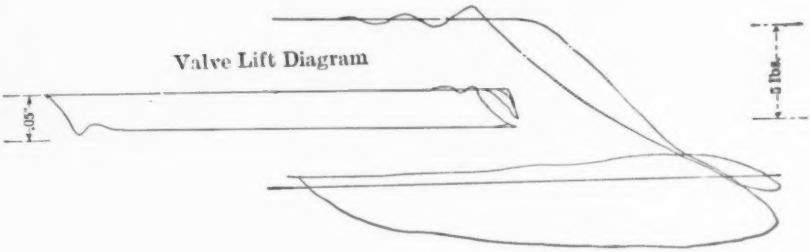


FIG. 25.

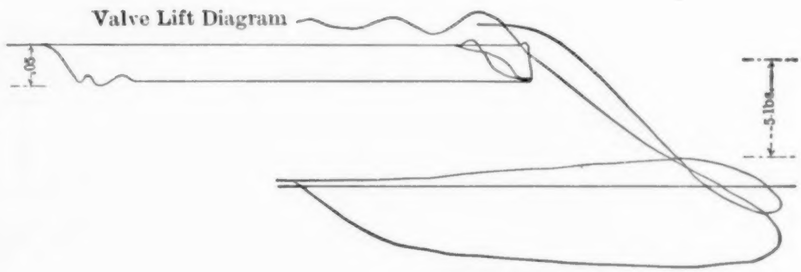


FIG. 26.

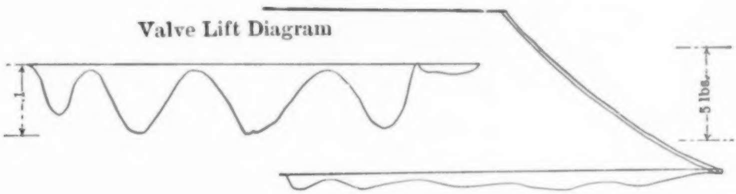
INTERMITTENT FLOW. (FLAT SEAT INLET VALVE. DAIMLER ENGINE.)

TABLE 22. RUN NO. 2 (A).

Maximum inlet valve lift 0.10. Average temperature of air = 77 degrees Fahr.

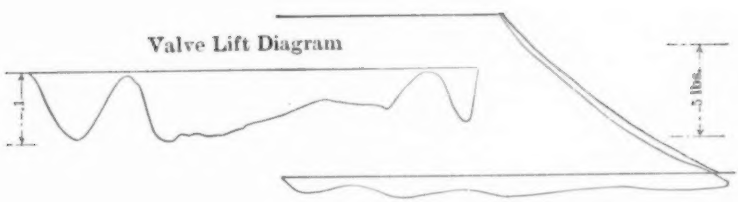
R. P. M.	Cubic Feet of Air pumped per Min.	Piston Displacement. (Cu. Ft. per Min.)	Per Cent. lost Volume.	Mean Resistance to Suction. lb. sq. inch.
150	2.60	2.94	8.65	.62
200	3.53	3.92	9.96	.92
250	4.49	4.90	8.37	1.
300	5.48	5.88	6.71	1.08
350	6.59	6.86	3.94	1.22
400	7.20	7.84	8.17	1.46
450	8.58	8.82	2.72	1.81
500	9.40	9.80	4.07	2.05
550	10.52	10.78	2.06	2.14

Valve lifts and pressure drop diagrams are shown in Figs. 27 to 36.



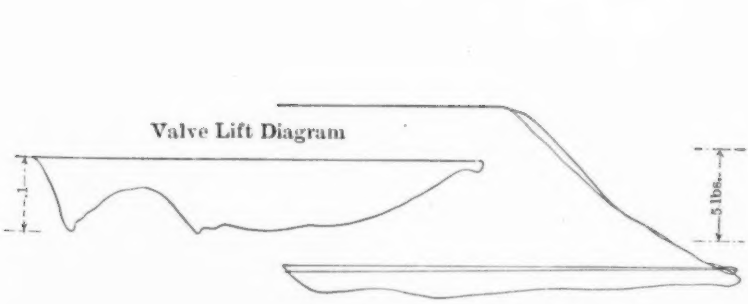
Pressure Drop Diagram
0.10-inch Max. Lift
150 R.P.M.

FIG. 27.



Pressure Drop Diagram
0.10-inch Max. Lift
200 R.P.M.

FIG. 28.



Pressure Drop Diagram
0.10-inch Max. Lift
250 R.P.M.

FIG. 29.

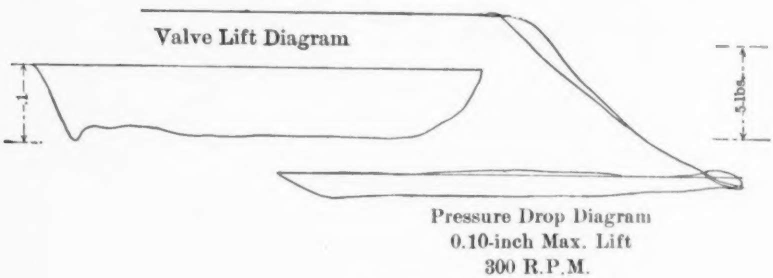


FIG. 30.

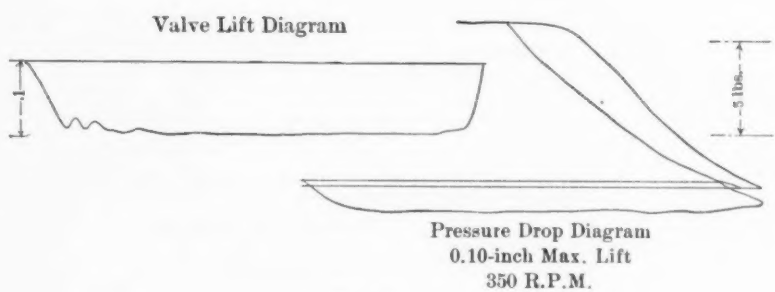


FIG. 31.

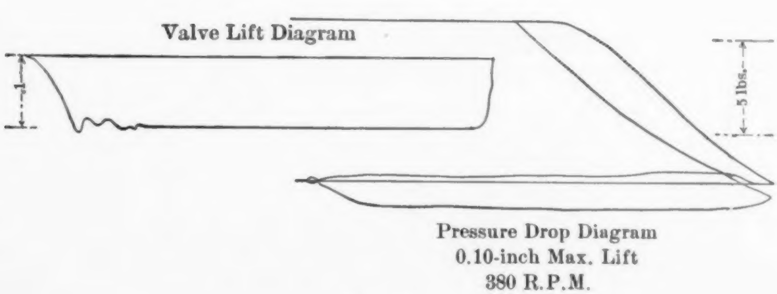
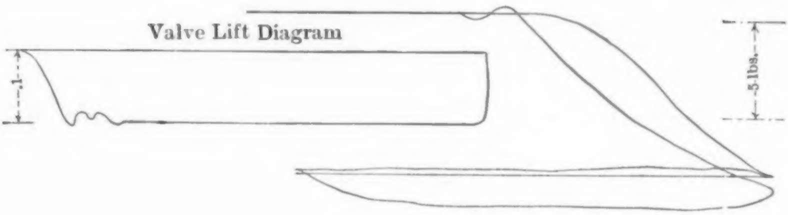
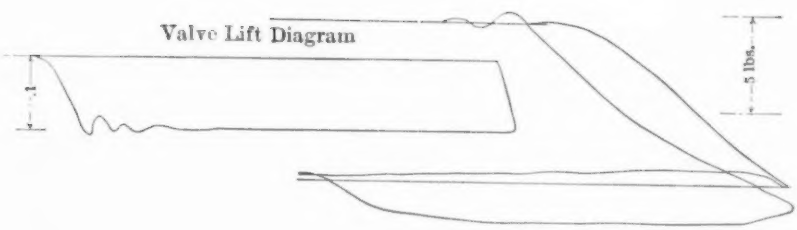


FIG. 32.



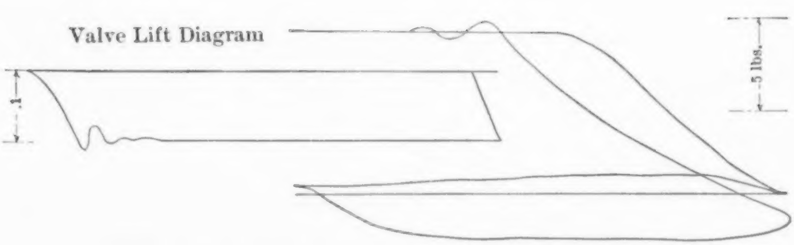
Pressure Drop Diagram
0.10-inch Max. Lift
400 R.P.M.

Fig. 33.



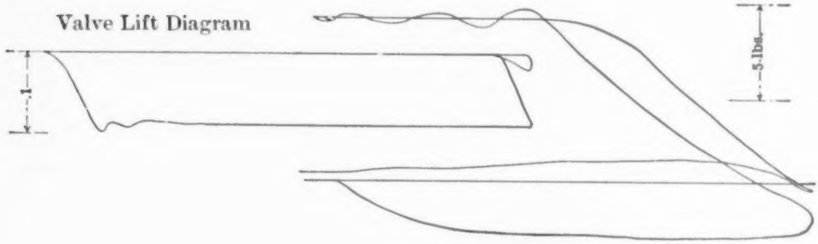
Pressure Drop Diagram
0.10-inch Max. Lift
450 R.P.M.

Fig. 34.



Pressure Drop Diagram
0.10-inch Max. Lift
500 R.P.M.

Fig. 35.



Pressure Drop Diagram
0.10-inch Max. Lift
565 R.P.M.

Fig. 36.

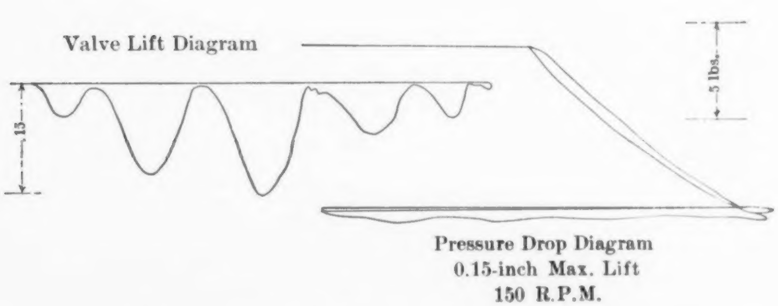
INTERMITTENT FLOW

Table 23. Run No. 3 (A). (Flat Seat Inlet Valve. Daimler Engine.)

Maximum inlet valve lift 0.15. Average temperature of air = 79 degrees Fahr.

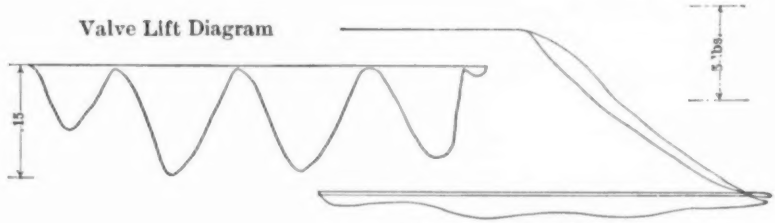
Speed R. P. M.	Cubic Feet of Air pumped per Min.	Piston Displacement. (Cu. Ft. per Min.)	Per Cent. lost Volume.	Mean Resistance to Suction. lb. sq. inch.
150	2.85	2.94	3.06	.68
200	2.79	3.92	3.32	.93
250	4.57	4.90	6.73	.96
300	5.60	5.88	4.76	1.08
350	6.45	6.86	5.98	1.15
400	7.54	7.84	3.83	1.29
450	8.26	8.82	6.44	1.35
500	9.50	9.80	3.06	1.55
560	10.38	10.98	5.47	2.05

Valve lift and pressure drop diagrams are shown in Figs. 37 to 45.



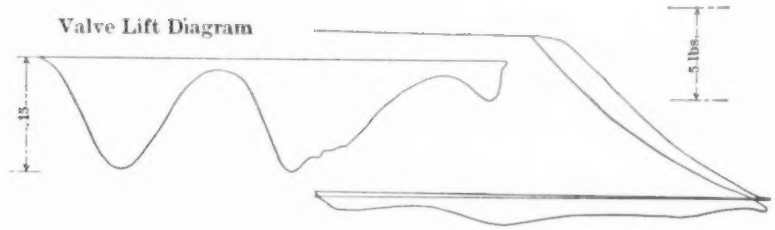
Pressure Drop Diagram
0.15-inch Max. Lift
150 R.P.M.

Fig. 37.



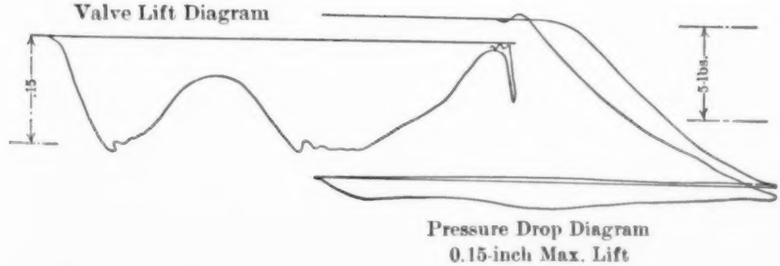
Pressure Drop Diagram
0.15-inch Max. Lift
200 R.P.M.

FIG. 38.



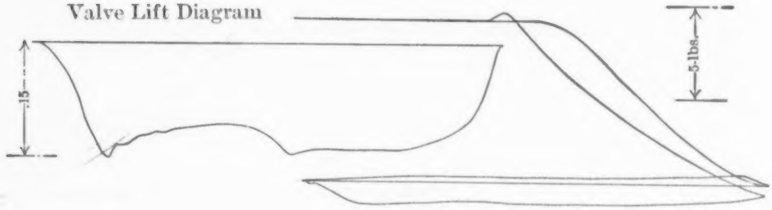
Pressure Drop Diagram
0.15-inch Max. Lift
250 R.P.M.

FIG. 39.



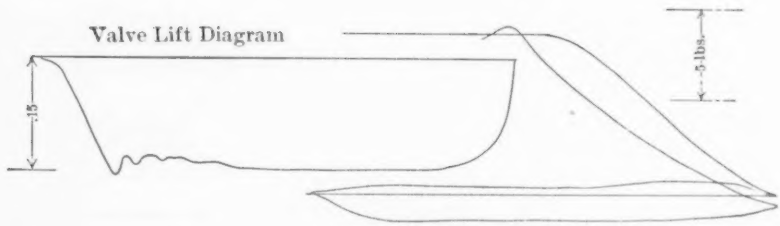
Pressure Drop Diagram
0.15-inch Max. Lift
300 R.P.M.

FIG. 40.



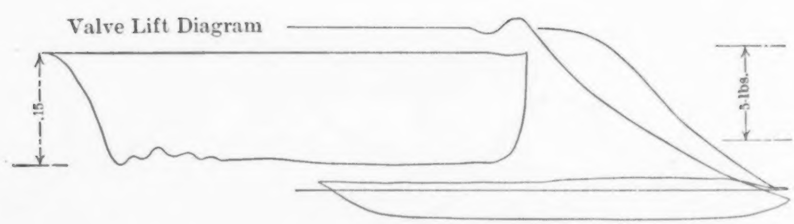
Pressure Drop Diagram
0.15-inch Max. Lift
350 R.P.M.

FIG. 41.



Pressure Drop Diagram
0.15-inch Max. Lift
400 R.P.M.

FIG. 42.



Pressure Drop Diagram
0.15-inch Max. Lift
450 R.P.M.

FIG. 43.

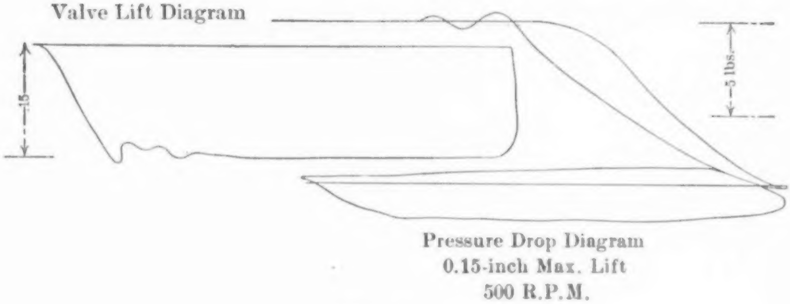


FIG. 44.

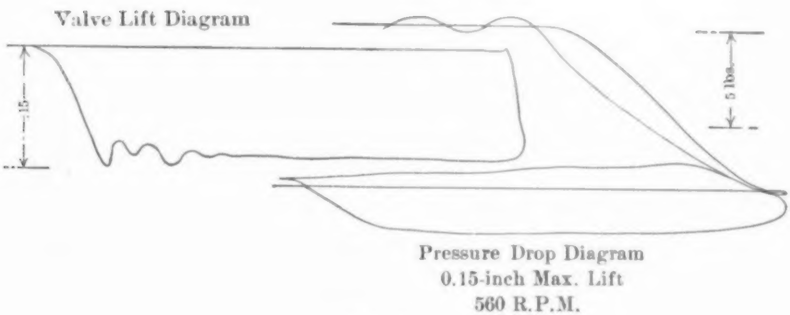


FIG. 45.

INTERMITTENT FLOW

TABLE 24. RUN NO. 4 (A). (FLAT SEAT INLET VALVE. DAIMLER ENGINE.)

Maximum inlet valve lift = 0.20. Average temperature of air = 81 degrees Fahr.

Speed R. P. M.	Cubic Feet of Air pumped per Min.	Piston Displacement.	Per Cent. lost Volume.	Mean Resistance to Suction. lb. sq. inch.
150	2.66	2.94	9.53	.55
200	3.56	3.92	9.18	.72
250	4.61	4.90	5.92	.93
300	5.55	5.88	5.61	.85
350	6.53	6.86	4.81	1.07
400	7.54	7.84	3.83	1.10
450	8.69	8.82	1.47	1.30
500	9.65	9.80	1.54	1.33
560	10.28	10.98	6.39	1.75

Valve lift diagrams and pressure drops are shown in Figs. 46 to 54.

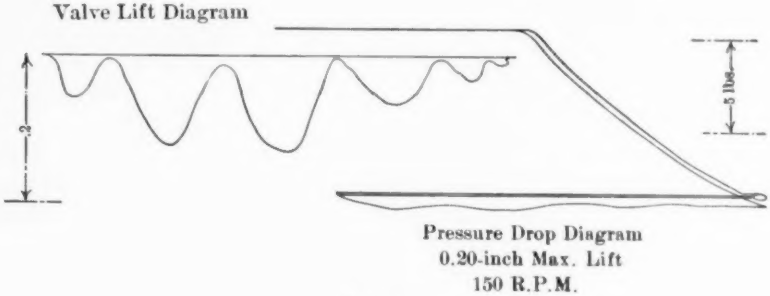


FIG. 46.

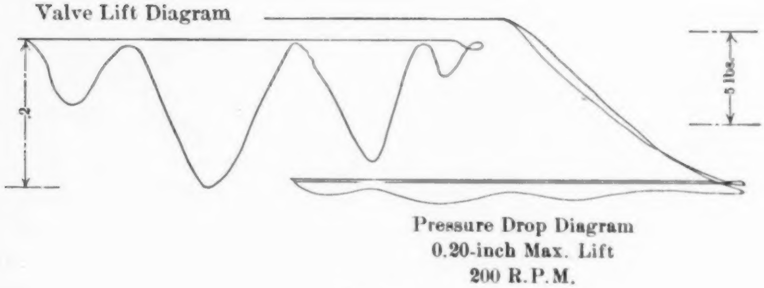


FIG. 47.

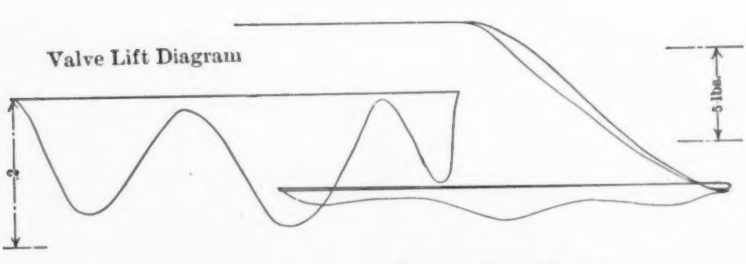
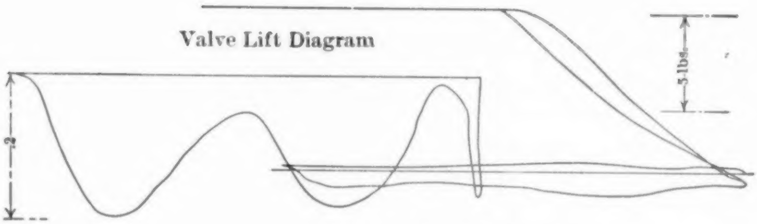
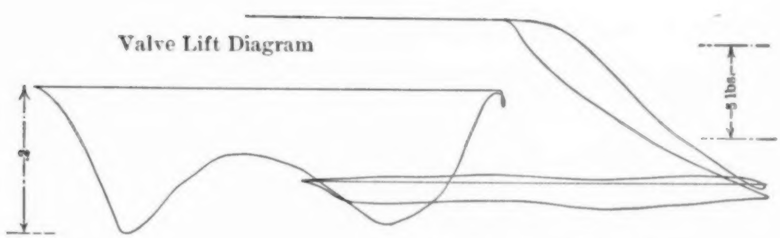


FIG. 48.



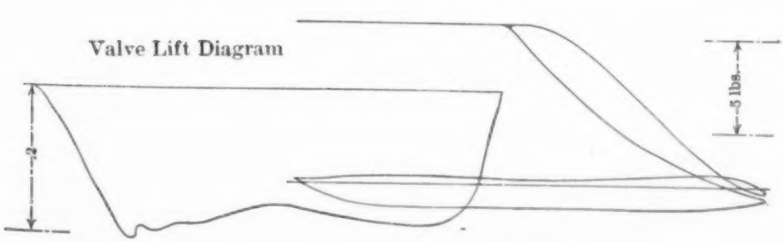
Pressure Drop Diagram
0.20-inch Max. Lift
300 R.P.M.

Fig. 49.



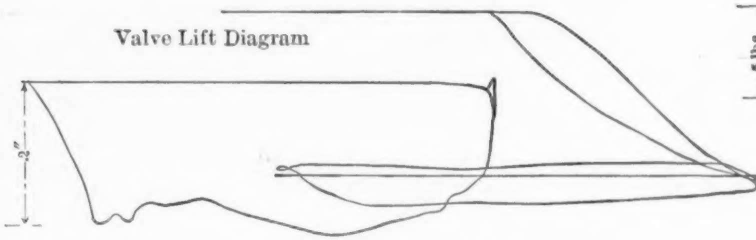
Pressure Drop Diagram
0.20-inch Max. Lift
350 R.P.M.

Fig. 50.



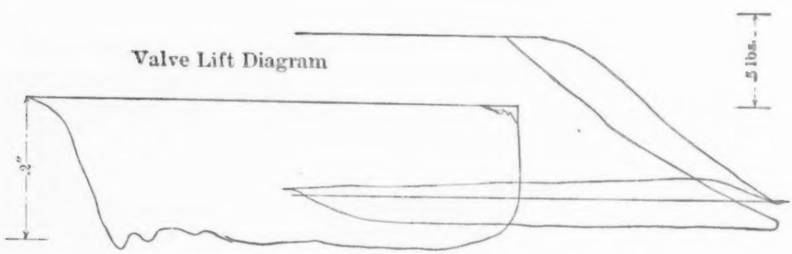
Pressure Drop Diagram
0.20-inch Max. Lift
400 R.P.M.

Fig. 51.



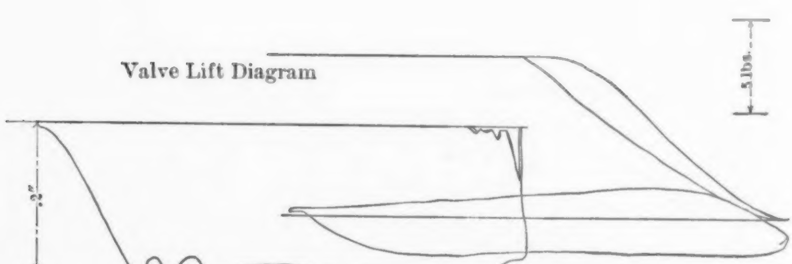
Pressure Drop Diagram
0.20-inch Max. Lift
450 R.P.M.

Fig. 52.



Pressure Drop Diagram
0.20-inch Max. Lift
500 R.P.M.

Fig. 53.



Pressure Drop Diagram
0.20-inch Max. Lift
560 R.P.M.

Fig. 54.

INTERMITTENT FLOW. (CONICAL SEAT EXHAUST VALVE. DAIMLER ENGINE.)

TABLE 25. RUN NO. 1 (B).

(1.25-inch conical seat valve; cam opened. Intermittent exhaust flow.)

Mean lift of exhaust value = 0.333 inch. Average temperature of air = 85 degrees Fahr.

Max. lift of inlet valve 0.10-in. Area valve opening 1.3 sq. inch.

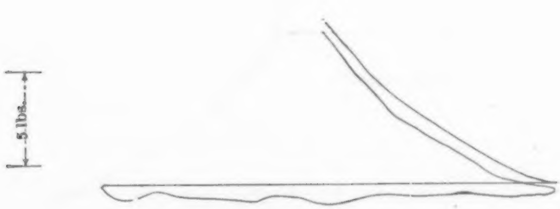
Max. lift of exhaust valve 0.40-in. Area of piston 12.177.

Speed R. P. M.	Cubic Feet of Air pumped per Minute.	Mean Resistance to Discharge. lb. sq. inch.	Terminal Discharge Pressure. lb. sq. inch.
150	2.83	...	...
200	3.69	...	...
250	4.63	...	about .1
300	5.56	.1	.20
350	6.54	.1	.2
400	7.56	.15	.3
450	8.50	.15	.3
500	9.39	.15	.3
570	10.41	.25	.5

Valve lift and pressure drop diagrams are shown in Figs. 55 to 64.

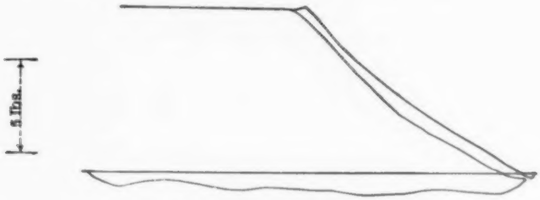
Daimler Exhaust Valve Lift Diagram
Valve Lift Curve
Begins at the Right Hand Side

FIG. 55.

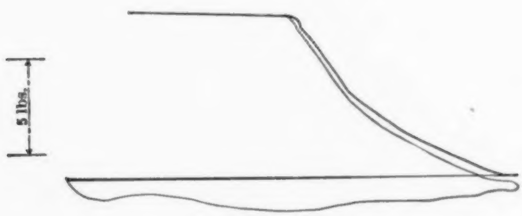


Pressure Drop Diagram
150 R.P.M.

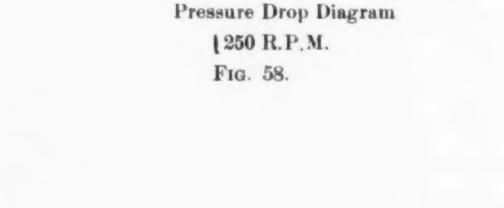
FIG. 56.



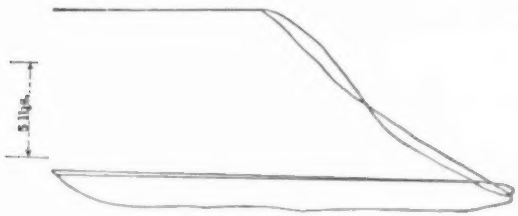
Pressure Drop Diagram
200 R.P.M.
FIG. 57.



Pressure Drop Diagram
250 R.P.M.
FIG. 58.

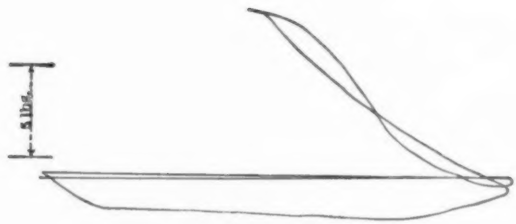


Pressure Drop Diagram
300 R.P.M.
FIG. 59.



Pressure Drop Diagram
350 R.P.M.

Fig. 60.

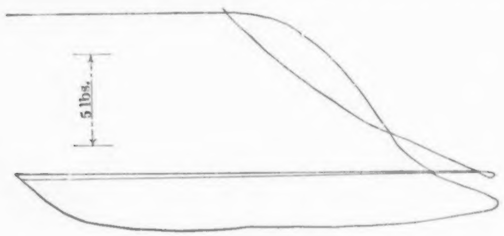


Pressure Drop Diagram
400 R.P.M.

Fig. 61.

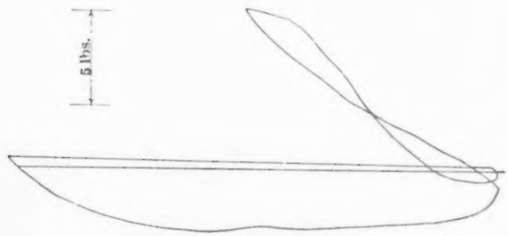
Pressure Drop Diagram
450 R.P.M.

Fig. 62.



Pressure Drop Diagram
500 R.P.M.

FIG. 63.



Pressure Drop Diagram
570 R.P.M.

FIG. 64.

INTERMITTENT FLOW. (CONICAL SEAT GAS VALVE. NASH ENGINE.)

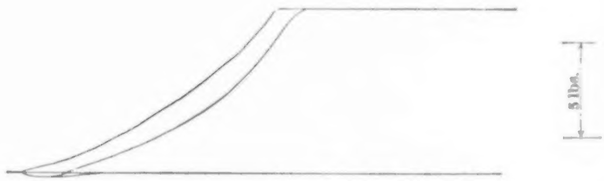
TABLE 26. RUN NO. 1 (A).

(1.5-inch conical seat valve, cam opened, intermittent inlet flow.)
Mean lift gas valve = 0.226 inch. Average temperature of air = 80 degrees Fahr.
Max. lift gas valve 0.34-in. Area piston = 33.18 sq. inch.
Area opening 1.23 sq. inch.

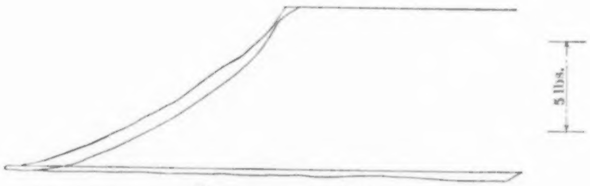
Speed R.P.M. (Average during Run.)	Cubic Feet of Air pumped per Min.	Piston Displacement. (Cu. Ft. per Min.)	Per Cent. lost Volume.	Mean Resistance to Suction. lb. sq. inch.
63	4.66	5.09	3.30	.36
96	8.73	9.22	5.32	.53
116	10.25	11.13	7.79	.64
140	12.23	13.43	9.00	.73
161	14.19	15.45	8.15	1.07
185	15.75	17.75	11.28	1.29
205	17.10	19.70	13.21	1.35
225	18.35	21.60	15.05	1.64
237	19.10	22.75	16.05	1.68
250	19.60	24.00	18.34	2.07
288	21.20	27.65	23.30	2.30
316	22.65	30.32	25.30	2.65

Valve lift and pressure drop diagrams are given in Figs. 65 to 80.

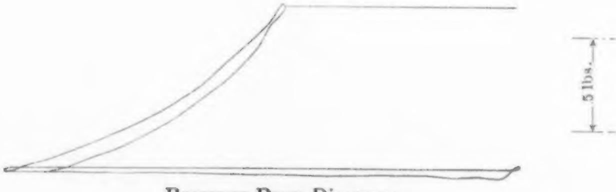
Nash Gas Valve Lift Diagram
Valve Lift Curve. Inlet Begins on Right Hand Side
FIG. 65.



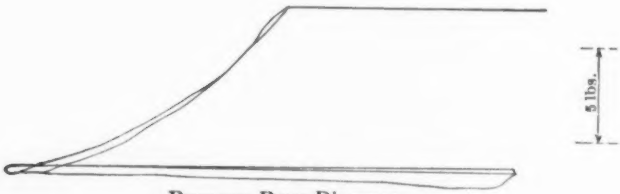
Pressure Drop Diagram
32 R.P.M.
FIG. 66.



Pressure Drop Diagram
55 R.P.M.
FIG. 67.



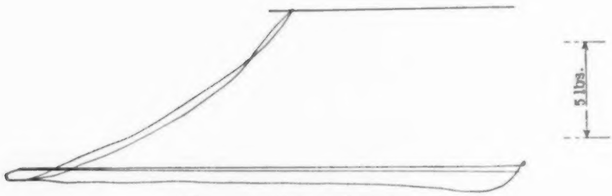
Pressure Drop Diagram
62 R.P.M.
FIG. 68.



Pressure Drop Diagram

95 R.P.M

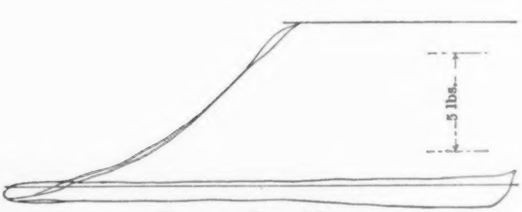
FIG. 69.



Pressure Drop Diagram

114 R.P.M.

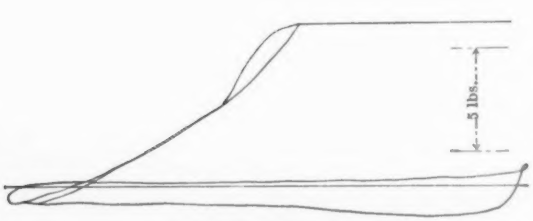
FIG. 70.



Pressure Drop Diagram

140 R.P.M.

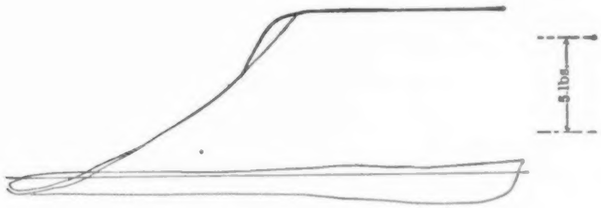
FIG. 71.



Pressure Drop Diagram

160 R.P.M.

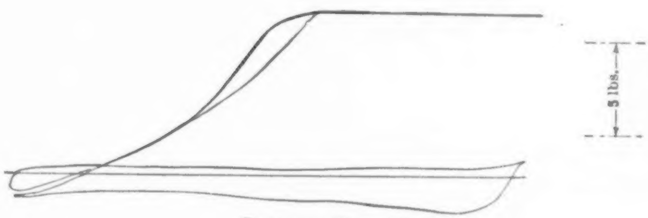
FIG. 72.



Pressure Drop Diagram

161 R.P.M.

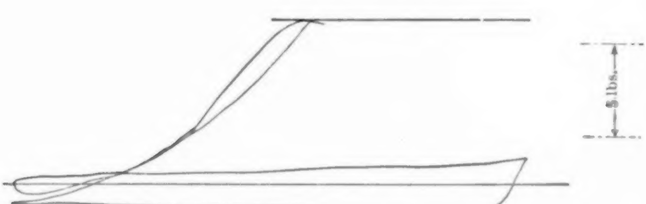
Fig. 73.



Pressure Drop Diagram

185 R.P.M

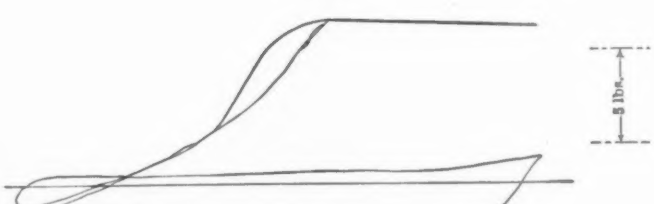
Fig. 74.



Pressure Drop Diagram

205 R.P.M.

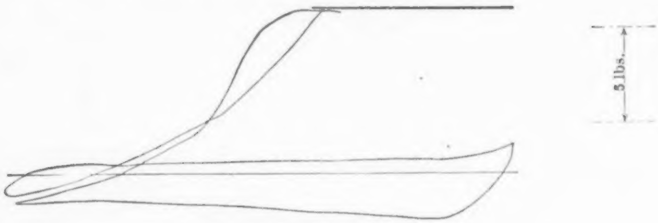
Fig. 75.



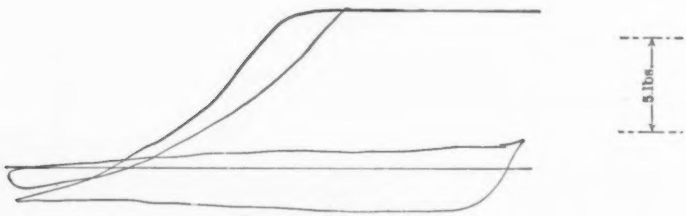
Pressure Drop Diagram

225 R.P.M.

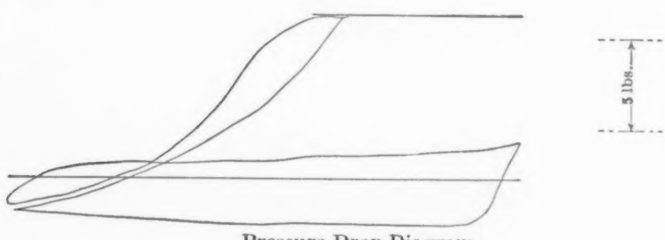
Fig. 76.



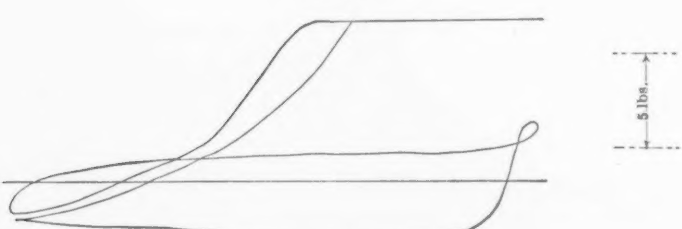
Pressure Drop Diagram
240 R.P.M.
FIG. 77.



Pressure Drop Diagram
250 R.P.M.
FIG. 78.



Pressure Drop Diagram
280 R.P.M.
FIG. 79.



Pressure Drop Diagram
312 R.P.M.
FIG. 80.

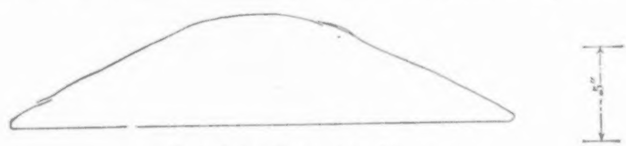
INTERMITTENT FLOW. CONICAL SEAT INLET VALVE.

TABLE 27. RUN No. 2 (A). (NASH ENGINE.)

(2-inch diameter conical seat valve, intermittent inlet flow.)
Mean lift of inlet valve = 0.393 inch. Average temperature of air = 80 deg. F.
Max. lift inlet valve 0.6-in.
Area of opening 2.97 sq. inches.

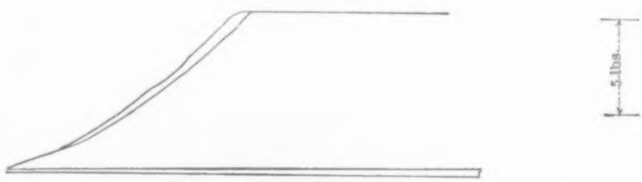
Speed R. P. M.	Cubic Feet of Air pumped per Min.	Piston Displacement. (Cu. Ft. per Min.)	Per Cent. lost Volume.	Mean Resistance to Suction.
102	9.46	9.79	5.32	...
138	12.53	13.23	5.29	...
176	16.21	16.90	4.08	.17
208	18.94	19.99	5.26	.25
247	22.10	23.72	6.84	.57
296	26.02	28.42	8.44	.71

Valve lift (Fig. 81) and pressure drop diagrams are given in Figs. 82 to 87.



Nash Inlet Valve Lift Diagram
Valve Lift Curve. Begins on Right Hand Side

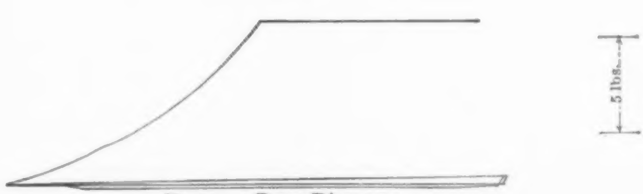
FIG. 81.



Pressure Drop Diagram

103 R.P.M.

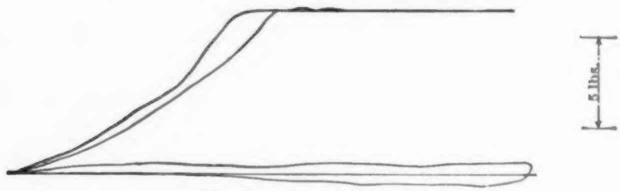
FIG. 82.



Pressure Drop Diagram

136 R.P.M.

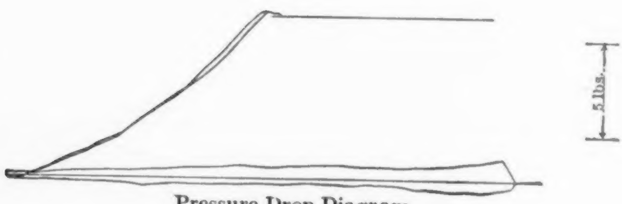
FIG. 83.



Pressure Drop Diagram

176 R.P.M.

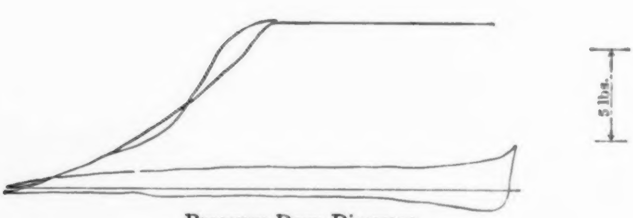
FIG. 84.



Pressure Drop Diagram

208 R.P.M.

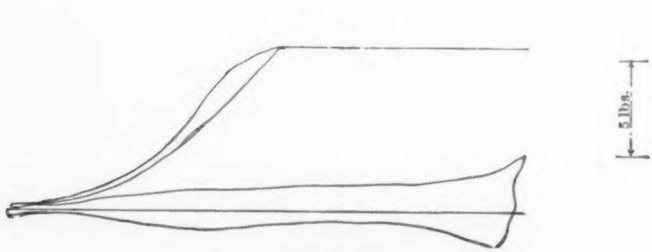
FIG. 85.



Pressure Drop Diagram

246 R.P.M.

FIG. 86.



Pressure Drop Diagram

291 R.P.M.

FIG. 87.

INTERMITTENT FLOW. CONICAL SEAT EXHAUST VALVE.

TABLE 28. RUN No. 1 (B). (NASH ENGINE.)

(2-inch diameter conical seat valve. Intermittent exhaust flow.)

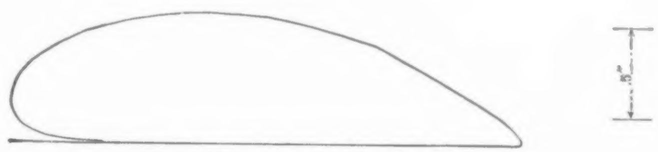
Mean lift of exhaust valve = 0.535 inch. Average temperature of air = 81 degrees Fahr.

Max. lift of exhaust valve 7-in.

Area of opening 3.61 sq. inches.

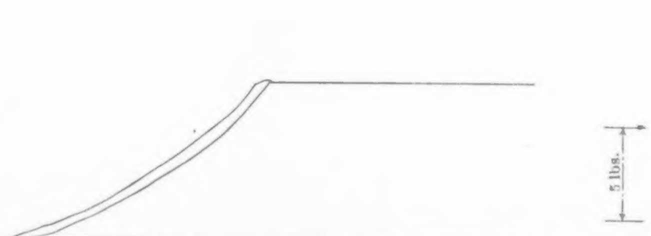
Speed R. P. M.	Cubic Feet of Air pumped per Minute.	Mean Resistance to Discharge.	Terminal Discharge Pressure.
80	7.6	...	...
117	10.80	...	...
151	13.80	.2	.8
185	16.50	.25	1.
210	18.25	.3	1.2
253	21.10	.4	2.4
283	23.00	.5	2.6
320	38.20	.6	3.

Valve lift (Fig. 88) and pressure drop diagrams are given in Figs. 87 to 96.



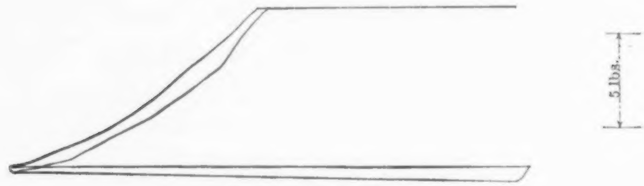
Exhaust Valve Lift Diagram
Nash Valve Lift Curve. Begins on Left Hand Side

FIG. 88.

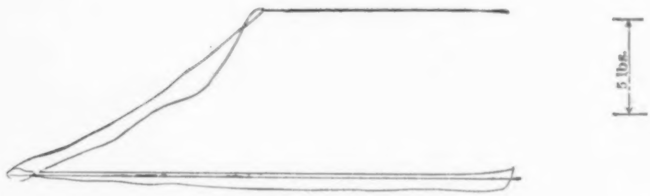


Pressure Drop Diagram
80 R.P.M.

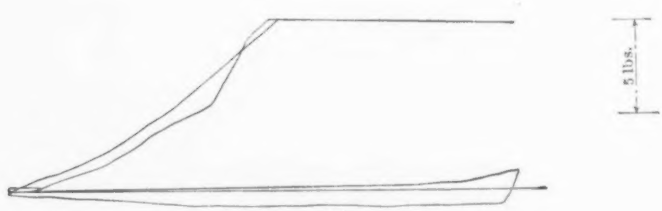
FIG. 89.



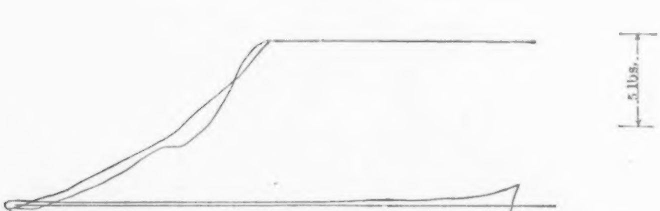
Pressure Drop Diagram
115 R.P.M.
FIG. 90.



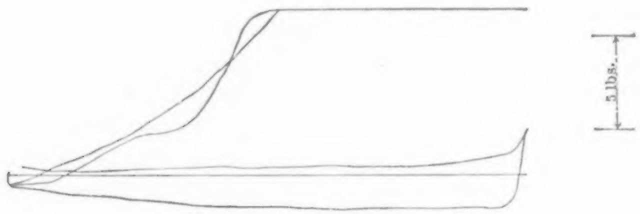
Pressure Drop Diagram
150 R.P.M.
FIG. 91.



Pressure Drop Diagram
183 R.P.M.
FIG. 92.



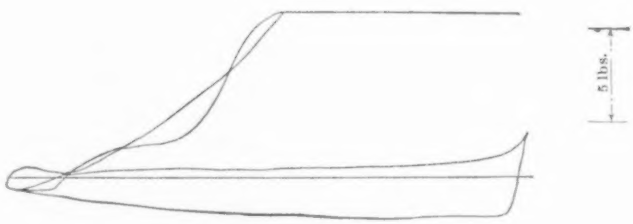
Pressure Drop Diagram
210 R.P.M.
FIG. 93.



Pressure Drop Diagram

252 R.P.M.

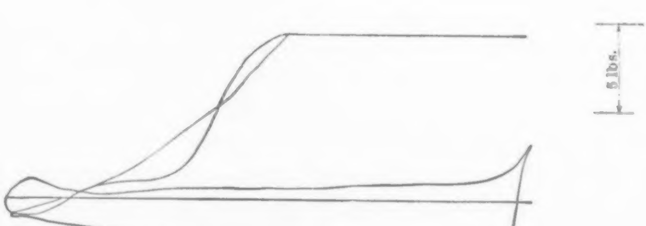
FIG. 94.



Pressure Drop Diagram

285 R.P.M.

FIG. 95.



Pressure Drop Diagram

320 R.P.M.

FIG. 96.

DISCUSSION.

Mr. Sanford A. Moss.—The work described in this paper is highly commendable and is just the sort of work which universities should be doing in large quantities for the benefit of the engineering fraternity. The experiments seem to have been intelligently planned and carefully executed. However, in order to make a paper such as this of any value some conclusion should be drawn, and it is to be hoped that Dr. Lucke may find time to summarize the results in revision.

One point has seemed to me evident from inspection of curves of Figs. 16 and 17, and that is that a conical seated valve is much more efficient than a flat seated one. Is this conclusion correct?

In the second part of the paper dealing with the intermittent flow there are given only the "Valve Lift" and "Pressure Drop" diagrams. Could not velocity co-efficients be obtained from these for comparison with the velocity co-efficients obtained in the first part of the paper? The device of stating the performance of a valve by means of a velocity co-efficient is very original and neat, and it seems to me should be used to the greatest extent possible.

Mr. Geo. Hill.—The writer is very glad to learn that it is intended to supplement this paper by presenting conclusions therefrom.

A consecutive numbering of the tables would facilitate reference to them.

An explanation of the more than perfect efficiency shown in a part of table, Run No. 2 (A), page 251, is desirable.

Do not the peculiar valve lift diagrams Figs. 18 to 54 inclusive indicate a defective mechanical condition (sticking of valve stem in guide or binding of spring on valve stem)?

Would not the results be affected by the varying size of the opening?

It is to be hoped that the author will show cuts of the cams that were employed, noting at what part of the stroke of the piston the cams become operative. It is suggested that the valve lift diagrams would be more readily understood if on the horizontal line the points where the piston reached the top and bottom of its travel respectively were noted. This information, in conjunction with the table in paragraph eight and the cam diagram, would make the results somewhat clearer.

Attention is called to the last line of table No. 1 on page 285; apparently there exists a typographical error.

Referring to the illustration Fig. 4: The sleeve through which the valve stem operates comes very close, apparently, to the cylindrical chamber, the end of which acts as the valve seat. No dimensions are given, but if the drawing is correct to scale there would apparently be a contraction of the airway at this point, which would seriously affect the results of all experiments made with this application.

Referring to Fig. 2: 1-inch pipe seems to be rather small—the design of this box would also seem to introduce an unnecessary number of angles, increasing the friction.

Referring to the tables for intermittent flow—tables on pages 261, 264, 268 and 271—it would appear that the most efficient lift for this particular valve was $\frac{2}{16}$ of an inch, and the most efficient speed 450 revolutions per minute. The maximum amount of air pumped per minute, however, corresponds with a lift of $\frac{1}{16}$ of an inch and a speed of 550 revolutions per minute. These tables do not appear to harmonize with themselves nor with one another in the matter of per cent. of volume lost, which one would naturally suppose would be a function of the lift of valve and speed in revolutions per minute. If the only obstructions to the flow of air were those due to the valve we should expect to see this followed out, but it seems hard to understand why the per cent. of volume lost (table, page 40) should be 9.53 at 150 revolutions, should drop to 3.83 at 400 revolutions, drop to 1.47 at 450 revolutions, rise to 1.54 at 500 revolutions, and then jump to 6.39 at 560 revolutions, while in the case of the table on page 264 the per cent. of volume lost varies from 9.96 at 200 revolutions to 2.06 at 550 revolutions; the valve lift diagram reflects some of this peculiarity, but the author does not explain why there should be produced such dissimilarity of diagrams as Figs. 27 and 31 evidence.

Comparing results given in the table on page 278 with the results given in the tables on page 264, it would appear that in this particular case the conical seat valve, while offering less resistance to suction in some way, caused a very much greater loss of volume of cubic feet pumped per minute, when compared with the flat-seat valve.

It is to be hoped that the author has more information to present in regard to the relative efficiency of the two forms of valves.

It would be of much interest to know what factor, if any, should be applied to these results to render them applicable to mixtures of gas and air and mixtures of gasolene vapor and air. It is generally accepted that they are by no means identical, and it is the writer's belief that results deduced from experiments made with air alone cannot be safely applied to the design of valves for gas or gasolene engines.

Recently published data indicates that the practical working of some valves may be very greatly improved by decreasing travel and changing the form of the operating cam.

During the past fourteen months the writer has for recreation been experimenting with a form of gasolene engine valve operating the engine of an automobile, gauging the results of the various changes by the speed obtained over a mile course on a country road which was always in practically the same condition when tests were made. These tests indicate in general that a velocity of the gases of 1,200 feet per second produce so much back pressure that the engine had nearly all it could do to run itself; that the delivered power increased with decrease of velocities to about 500 feet per second; that there was no appreciable increase in delivered power when the velocities were reduced to 250 feet per second, the velocities in all cases being figured from the area of the valve opening, the time during which it was opened, and the piston displacement, and no allowance was made for any expansion of the exhaust charge, which must have expanded materially and must have therefore produced higher velocities than those mentioned.

Various forms of cams were used and the best results were obtained from those in which the lifting portion was a straight line tangent to the concentric portion of the cam, the valve lifter traveling on a second concentric portion as soon as the maximum lift was obtained—thus for the exhaust portion of the cam the lift began when the crank still had 30 degrees of arc to pass over before reaching the bottom center. The valve reached its full open position when the crank had passed the bottom center by 14 degrees. It began to close when the crank still had 30 degrees of arc to travel, and closed entirely when the crank had passed the top center by 10 degrees. Translated into piston movement for a 4-inch stroke, this meant that release occurred when the piston still had about ¼ of an inch to travel; that during all of the upstroke of the piston, except the last quarter inch, the

valve remained wide open; that while the piston was traveling the last quarter inch of its stroke the valve was closing—closing completely before the piston had traveled downward $\frac{1}{32}$ of an inch.

Prof. F. R. Hutton.—In presenting this paper on behalf of the author, I would like to call attention to the practical significance of these investigations which the author has outlined in a concise way in his first paragraph.

The internal combustion engine receives its fuel supply to the cylinder, either under no pressure at all or under a very light one. Hence the effect of inertia in the mass of mixture of fluid and air, and the effect of inertia and fluid friction at the valve itself, act as the speed increases to diminish the effective weight of heat energy which actually enters the cylinder. This is roughly stated by placing as one of the limitations of the internal combustion engine, the principle that the power developed by the engine does not increase directly or proportionately to the number of revolutions as is the case with the steam engine where the working medium is delivered under pressure through the valves, and if the ports have any intelligent area whatever, the loss will not increase with speed. Up to a certain speed, say between 1,200 and 1,600 revolutions per minute, the output is quite closely proportional to the speed. Beyond that the increase of power is not as the increase of speed.

Mr. Geo. Hill.—Professor Hutton made the statement that the power developed does not increase with the speed. It may not, of course, directly with the speed, but from my observation it does increase very nearly with the revolutions and the speed, and up to 1,600 revolutions per minute the output appears to be quite closely proportioned to the speed.

Professor Hutton.—I should have restricted my statement. After a certain speed is reached the increase is not as the speed. It does increase up to a certain limit; beyond that the speed is not increased as the power—while the speed increases it does not increase as the power.

Mr. J. C. Parker.—About twelve years ago I demonstrated by experiment that throttling the charge by reduction of the opening through the poppet valves of a gas engine, if accompanied by a corresponding reduction of the compression space, would lead to improved economy, but at some expense of efficiency.

*Prof. Chas. E. Lucke.**—The appreciation expressed by Mr.

* Author's Closure, under the Rules.

S. A. Moss is very gratifying, coming as it does after the execution of such a large quantity of work. The only reason conclusions were not inserted in the original paper was lack of time. They are now presented, but the conclusions presented may not be those that some of the readers of the paper might expect. It will be impossible, as Mr. Moss, for instance, desires, to state which type of valve is the better. The comparisons made by him between Figs. 16 and 17 are not quite fair, because for equal lifts the areas of the openings are not the same for conical seat valves and flat seat valves. The opening for a flat seat valve is always larger than for a conical seat valve of the same diameter and same lift, and one of the things brought out by the steady flow curves is that the co-efficient is dependent as much upon opening as it is upon anything else.

In accordance with the suggestion of Mr. Hill the tables are renumbered consecutively. The more than perfect efficiency shown on page 251 may be explained in three ways. There may have been an error in calculating the valve opening, as has already been pointed out; there may have also been an error in the meter reading, and, what is more likely, it may be improper to take air at meter density. I am rather inclined to believe that the last is the true explanation. Mr. Hill's question concerning the effect of varying the size of the valve must be answered in the affirmative. The size will affect the results, and this is shown by comparing the results of the different size valves used. It is not necessary in this work to give the cam curves, as asked for by Mr. Hill, because the valve lift curves are given on the same stroke base as the indicator diagrams. The only effect of the cam is the lift of the valve, and the actual effect is shown in the valve lift curve, which is very much more satisfactory than the cam curve, and likewise more exact. There seems to be no error in the tables on page 285. The illustration in Fig. 4 is approximately to scale, and while there may be some contraction of the airway it is not very serious, because the area at the point of greatest contraction is about four times the maximum valve area used. The friction of the 1″ pipe, Fig. 2, is appreciable, and Mr. Hill's question on this point can be answered by comparing the runs made with and without housing and marked Run A1 and Run A2. Drawing conclusions of the efficiency of the valve with respect to meter readings which give the air pumped is not justified when successive values differ by 3 per cent. or less as the meter may be

this much in error. Tables on the quantity of air pumped are inconsistent and the cause is likely in the meter which has an error and which introduces pressure. The results tabulated are given exactly as found and are not corrected because it was deemed important that the limit of accuracy of the work should be known to the readers of the paper. Every possible precaution was taken and the results reported without correction. The comparison by Mr. Hill of the table on page 278 with that on page 264 is a little unfair because the velocity is different through two valves for equal lifts, one being a flat seat and the other a conical seat valve.

These experiments were undertaken not so much to prove the superiority of one type of valve over the other as to provide some definite data on which the designer may predict the results that will be given by the valve he selects. The selection of the type of valve is governed rather by structural reasons than by efficiency of flow through it. The type of valve being fixed, the designer must approximate the suction and exhaust lines of the indicator cards to determine the mean resistance and the mechanical work lost in charging and discharging the gases, also the amount of charge lost by valve resistance and the amount of old charge retained from the same cause. The charge which is lost during suction is measured at both ends of the atmospheric line where the suction line and compression line cross it. That part which is lost at the beginning of the stroke cannot be properly laid to an improper valve size or lift because it is more a matter of valve or cam adjustment and terminal exhaust pressure in gas engines than valve opening. To be sure, automatic valves must first be opened by the pressure difference against the spring resistance and so cause a loss, but in this case the suction line before the valve opens is merely the expansion line for the clearance volume and is easily predicted. The part of the charge lost at the end of the suction stroke, and which is shown by the crossing of the compression and atmospheric lines, is also as much a matter of valve setting as it is of valve opening. If the valve be held open for a short period after dead center most of this loss will be recovered. The indicator cards shown for cam lifted valves proves this. This correction by setting cannot be made for automatic valves, but then the valve stays open after the end of the stroke until the pressure difference falls to the equivalent of the spring tension, when the charge will be compressed. This beginning of compression, how-

ever, with automatic valves does not occur at the end of the stroke as appears from the indicator cards. On these cards the pressure difference equivalent to the spring tension is 0.8 pounds per square inch. There is no way known to the author in which this point can be determined on the indicator cards, and no data can be given so simple as to be practically valuable. It was for this reason that the large number of cards was presented, each giving the actual curve with the exact conditions under which it was determined and from which it resulted.

The characteristic of the suction and discharge lines at points where the flow is increasing or decreasing is indeterminate and can only be guessed at, for besides the above influences the question of gas inertia, pipe friction and flow enter. With the set of cards presented in this paper, however, the designer should be able to make a good guess. There is one point on each of the lines that one should be able to determine with fair certainty, and this is the suction vacuum or discharge pressure at the point of zero acceleration for the piston and gases. This is a point where the piston has a uniform motion and occurs at 45 per cent. of the instroke and 55 per cent. of the outstroke for this ratio of connecting rod to crank. The pressure drop at this point of the stroke is due solely to valve opening and piston speed and is not affected by inertia of the gas, nor is it subject to modification by any valve adjustments. At this point the values for the co-efficient of efflux determined from any experimental data should apply practically.

The only data before the designer are his valve opening and piston speed. These are both a maximum at the point of zero acceleration or uniform velocity. If the rate of displacement of the piston at this point be taken as the number of cubic feet of flow at atmospheric pressure and this volume be divided by the maximum area of the valve opening, there will result a certain velocity. This is ordinarily termed the gas velocity in the process of designing. With this velocity there should occur a certain pressure drop, and the pressure drop that will occur is greater actually than would be calculated by the square root formula. How much greater it will be is shown by the co-efficients which are calculated for the intermittent flows and given below. To the right and left of this point of the diagram the pressure drop will be greater or less, depending upon whether the ratio of valve opening to actual piston speeds remains the same as at the point of uniform velocity or not. This is easily found out by comparing

the valve lift curve with the piston speed curve, as explained in books on design.

The following Table 29 is determined from the intermittent flow by the indicator cards for the Daimler exhaust valve, and shows the speeds, pressure drops, gas velocity calculated on piston speed and valve opening, and the co-efficient of efflux. Velocities are in feet per minute.

TABLE 29.

DAIMLER EXHAUST VALVE.

Conical Seat, Cam Opened.

Speeds R. P. M.	Press. drop from cards, lb. sq. inch	Speeds	Press. drop from curve	Piston speeds	Calcul. gas vel.	Theoret. velocity	Co-efficient of efflux
150	...	150	.015	218.4	2,000	2,650	.75
200	....	200	.037	291.2	2,700	3,720	.725
250	.05	250	.05	364.0	3,400	4,750	.715
300	.10	300	.075	436.0	4,000	5,850	.685
350	.12	350	.11	509.6	4,750	6,850	.695
400	.15	400	.15	580.4	5,425	8,300	.655
450	.17	450	.20	653.2	6,100	9,600	.635
500	.20	500	.26	726.0	6,800	11,000	.62
570	.40	550	.35	798.8	7,500	12,600	.595

From the above figures and the curves, Figs. 97 and 98, it appears that the co-efficients of efflux at the point of zero acceleration

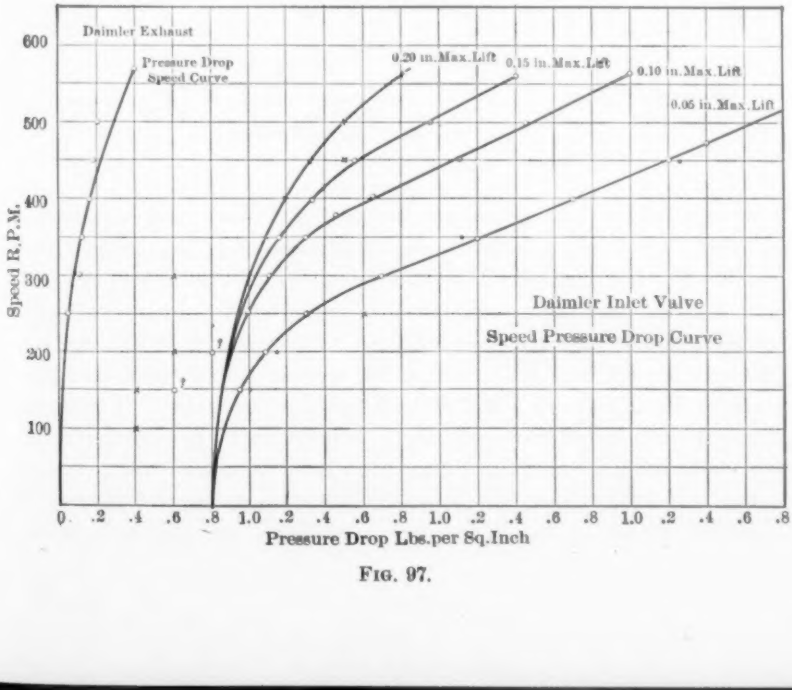


FIG. 97.

for this cam lifted conical seat exhaust valve decrease slightly with pressure drop—that is, decrease as flow increases, the two limits for the co-efficients being 0.75 and 0.60 when the velocities as calculated vary from 2,000 to 7,500 feet per minute. Table 30 gives the same information as the preceding one for one of the flat seat valves, which are automatic in type and which have a spring tension of 0.8 pounds per square inch of valve. This table gives co-efficients for four different maximum lifts.

It appears from the above that the co-efficient of efflux for intermittent flow through these automatic flat seat valves increase with

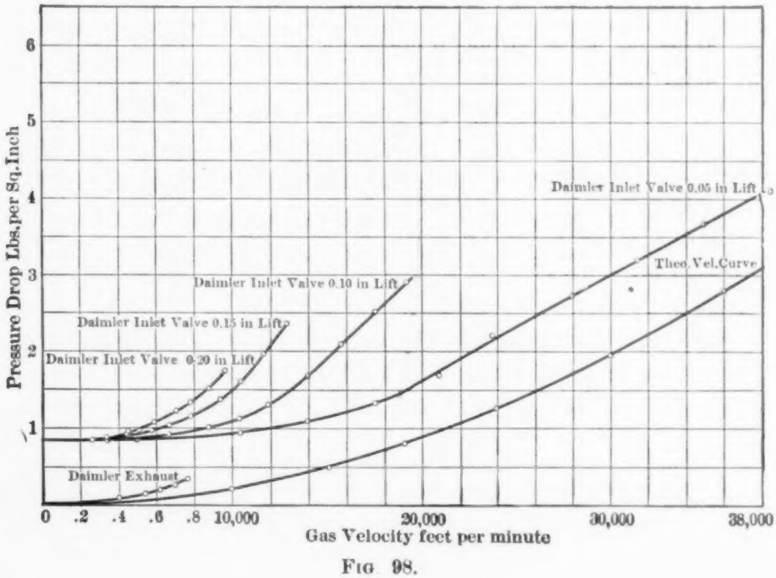


FIG. 98.

pressure drop—that is, increase with velocity and speed, which is a different result than was obtained for the conical seat cam lifted valve. For the smallest lift the co-efficients lie between 50 per cent. and 89 per cent., for corresponding calculated velocity between 10,000 and 38,000 feet per minute. For twice this lift the co-efficients lie between 0.2 and 0.5, the corresponding calculated gas velocity being between 5,000 and 19,000 feet per minute. For three times the minimum the co-efficients lie between 17 per cent. and 39 per cent. when the corresponding calculated velocities are between 3,400 and 12,000 feet per minute. For the maximum lift, which was four times the minimum, the co-efficients are still less, lying

between 13 per cent. and 34 per cent. when the calculated velocities lie between 2,600 and 9,600 feet per minute.

Calculated in a similar way from data on the Nash gas valve

TABLE 30.

DAIMLER ENGINE INLET VALVE.

Flat Seat, Automatic.

A. 0.05 inch lift.

Tension of spring = 0.8 pound. $\frac{\text{Piston area}}{\text{valve area}} = 48$

Speeds R. P. M.	Press. drop from cards, lb. sq. inch	Speeds	Press. drop from curve	Piston speeds	Calcul. gas vel.	Theoret. velocity	Co-efficient of efflux
150	.95	150	.95	218.4	10,460	21,000	.50
200	1.1	200	1.10	291.2	13,970	22,500	.62
250	1.3	250	1.30	364.0	17,470	24,500	.71
300	1.7	300	1.70	436.8	20,920	28,000	.75
350	2.2	350	2.20	509.6	27,730	32,000	.73
400	2.7	400	2.70	582.4	28,000	35,500	.79
450	3.2	450	3.20	654.2	31,400	39,000	.80
475	3.4	500	3.65	728.0	34,940	41,000	.85
575	4.4	550	4.10	798.8	38,400	43,250	.89

B. 0.10 inch lift.

$\frac{\text{Piston area}}{\text{valve area}} = 24$

Speeds R.P.M.	Press. drop from cards, lb. sq. inch	Speeds	Press. drop from curve	Piston speeds	Calcul. gas vel.	Theoret. velocity	Co-efficient of efflux
150	.6	150	.86	218. 4	5,230	20,000	.26
200	.8	200	.92	291.2	6,635	20,500	.32
250	.1	250	.1	364.0	8,735	21,500	.4
300	1.1	300	1.1	436.8	10,460	22,500	.46
350	1.3	350	1.3	509.6	11,865	24,500	.48
380	1.45	...	....	......	......	......	...
400	1.65	400	1.65	582.4	14,000	27,500	.5
450	2.1	450	2.08	654.2	15,680	31,000	.505
500	2.45	500	2.5	728.0	17,470	34,000	.515
565	3.	550	2.9	798.8	19,100	36,751	.52

C. 1.5 inch lift.

$$\frac{\text{Piston area}}{\text{valve area}}=16$$

Speeds R.P.M.	Press. drop from cards, lb. sq. inch	Speeds	Press. drop from curve	Piston speeds	Calcul. gas vel.	Theoret. velocity	Co-efficient of efflux
350	1.15	150	.85	218.4	3,486	20,000	.174
400	1.35	200	.9	291.2	4,423	20,500	.215
450	1.55	250	.96	364.0	5,825	21,000	.277
500	1.95	300	1.04	436.8	6,640	22,000	.302
560	2.4	350	1.15	509.6	7,910	23,000	.344
...	...	400	1.35	582.4	9,320	25,000	.373
...	...	450	1.55	654.2	10,460	26,750	.380
...	...	500	1.95	728.0	11,646	30,000	.39 –
...	...	550	2.32	798.8	12,800	32,750	.39 +

D. 0.20 inch lift.

$$\frac{\text{Piston area}}{\text{valve area}}=12$$

Speeds R.P.M.	Press. drop from cards, lb. sq. inch.	Speed	Press. drop from curve	Piston speeds	Calcul. gas vel.	Theoret. velocity	Co-efficient of efflux
150	.40	150	.85	218.4	2,615	20,000	.13
200	.60	200	.89	291.2	3,317	20,205	.164
250	1.60	250	.94	364.0	4,367	20,750	.2
300	.6	300	1.0	436.8	5,230	21,500	.243
350	1.1	350	1.1 –	509.6	5,932	22,500	.263
400	1.25	400	1.2	582.4	7,000	23,500	.31
450	1.5	450	1.33	654.2	7,840	24,750	.32
500	1.5	500	1.5	728.0	8,735	26,250	.33
560	1.8	550	1.75	798.8	9,600	28,500	.34

in Table 31 will be found information on co-efficients with the conditions under which they were determined.

From this table it appears that the co-efficients increase slightly with pressure drop and the rate of flow, lying between 52 per cent. and 56 per cent. when the calculated gas velocity lies between 3,500 and 21,000 feet per minute, the nature of the variation being quite the same as for the steady flows reported in the early part

TABLE 31.

NASH GAS VALVE.

Conical Seat, Cam Opened.

Area of opening at maximum lift=1.227 square inch.
Area of piston=33.185 square inches.
$\frac{\text{Area of piston}}{\text{Area of valve opening}}=27$

Speeds R.P.M.	Press. drop from cards, lb. sq. inch	Piston speeds	Calcul. Gas vel.	Theoret. velocity	Co-efficient of efflux
50	.10	130.9	3,530	6,800	.52
100	.40	261.8	7,060	13,500	.52
150	.80	392.7	10,670	19,000	.56
200	1.40	523.6	14,130	25,500	.55
250	2.10	654.5	17,670	31,000	.565
300	2.95	785.4	21,220	37,500	.565

of the paper. The Nash inlet valve, which is the same in type as the Nash gas valve, gives the co-efficients and velocities reported in Table No. 32.

These co-efficients lie between 51 and 47 and decrease with flow slightly. The corresponding gas velocity calculated from the piston displacement for these two limits are 3,000 and 9,000 feet per minute. The fact that co-efficients for these two Nash valves with conical seats and inlet flow do not vary very much is quite interest-

TABLE 32.

NASH INLET VALVE.

Conical Seat, Cam Opened.

Maximum lift=0.6 inch.
Area of opening at maximum lift=2.97 square inches.
Area of piston=33.18 square inches.
$\frac{\text{Area of piston}}{\text{Area of valve opening}}=11.05$

Speeds R.P.M.	Press. drop from curves, lb. sq. inch.	Piston speeds	Calcul. Gas veloc.	Theoret. velocity	Co-efficient of efflux
50	...	.....	.....	....	....
100	.08	261.8	3,100	6,100	.51
150	.18	392.7	4,520	9,100	.50
200	.33	523.6	6,020	12,300	.495
250	.54	654.5	7,520	15,750	.48
300	.80	785.4	9,040	19,000	.475

ing. It also appears since one increases and the other decreases that the influence of the chamber in which the valve is housed has an appreciable effect upon the contraction. Another odd situation appears in the comparison of the preceding results for inlet to the following results in Table No. 33 for exhaust flow.

TABLE 33.

NASH EXHAUST VALVE.

Conical Seat, Cam Opened.

Maximum lift=0.7 inch.
Area of opening at maximum lift=3.61 square inches.

$$\frac{\text{Area of piston}}{\text{Area of valve opening}}=9.19$$

Speeds R.P.M.	Press. drop from curves, lb. sq. inch	Piston speeds	Calcul. Gas veloc.	Theoret. velocity	Co-efficient of efflux
100	.05	261.8	2,400	4,750	.505
150	.12	392.7	3,610	7,400	.487
200	.23	523.6	4,820	10,250	.47
250	.38	654.5	6,010	13,100	.46
300	.60	785.4	7,220	16,500	.437

For the exhaust flow the co-efficients vary between 51 per cent. and 44 per cent. for gas velocity calculated on piston displace-

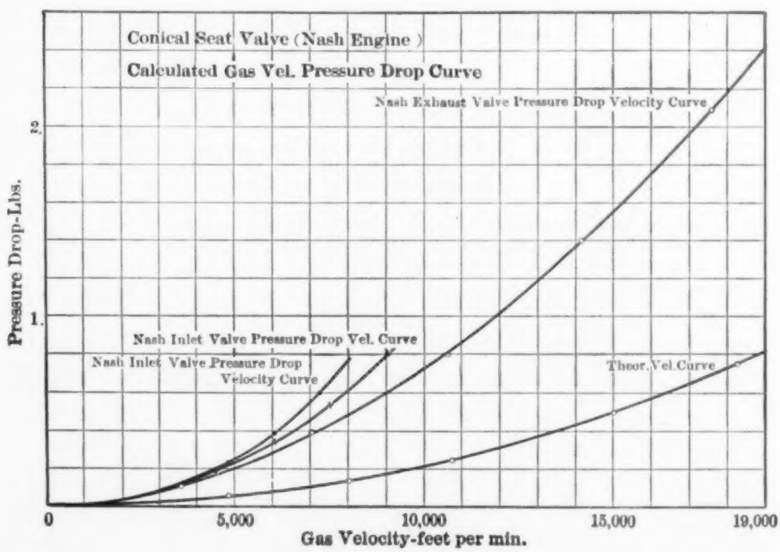


FIG. 99.

ments between 2,000 and 7,000 feet per minute. In every case the co-efficients are a little less than for inlet valve flow, but not nearly as much as might be expected. In the set of curves, Fig. 99, the tabular results are shown graphically. From this table the pressure drop can be read off at once as in Figs. 97 and 98 for the Daimler engine, for any given gas velocity in feet per minute, calculated on piston displacement at the point of uniform motion.

It might seem as if the co-efficient of efflux for steady flow should be applicable to intermittent flows at this point of zero acceleration, but the preceding results prove that this is not the case. The actual co-efficients, as determined from the indicator cards, are smaller for the intermittent flow, even though the velocity is uniform at the point in question. The form in which these results are finally given make them available for design work.

No. 1100.*

RESULTS OF THE PRELIMINARY PRODUCER GAS TESTS OF THE UNITED STATES GEOLOGICAL SURVEY COAL-TESTING PLANT AT ST. LOUIS.†

BY R. H. FERNALD, ST. LOUIS, MO.
(Member of the Society.)

1. An obviously necessary regulation of the Government prohibits the publication of results of the United States Geological Survey Coal Testing Plant until such results have appeared in official Government reports. This regulation is in a way unfortunate, as it prohibits the presentation at this meeting of the most reliable and most efficient results of the producer gas tests. It not only necessitates confining the information and details to results already published by the United States Geological Survey,‡ but forces the writer to confine his attention to preliminary work conducted under the erratic and often exasperating conditions of an Exposition period.

2. The operating conditions that are now maintained in the Gas Producer Division of the Testing Plant are so superior to those that were possible during the preliminary period that the results of the past several months show marked improvement over the figures here presented, but, owing to the Government restriction, they can be mentioned in a general way only at this time.

Equipment.

3. The plant upon which these tests were made is a Taylor pressure gas producer, furnished by R. D. Wood & Co., of Philadelphia. It is designated as a 250 horse-power producer gas power plant. The cut, Fig. 1, presents the general arrangement of

* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

† Presented by permission of the Director of the United States Geological Survey.

‡ For detailed description of the Plant and complete reports see Bulletin No. 261 of the United States Geological Survey and other reports to be published upon this subject.

the plant, with the boiler house (erected in connection with the boiler and steam engine trials and not directly connected with the producer gas plant)—directly in the rear. At the extreme right is the engine room. The exhaust pipe of the gas engine is seen projecting through the roof.

4. In Figs. 2 and 3 are shown the plans and elevations of the different units of the plant, as well as the general dimensions. The specifications given in the catalogue, published by R. D. Wood & Co., for the producer are: "Gas producer with rotative ash table; design *A*; size, No. 7; inside diameter of brick lining or jacket, 7 feet; area of fuel bed, 38.5 square feet; height of top casing, 15 feet."

(A complete detailed description of the entire plant will appear in the report to be published by the United States Geological Survey in a few weeks.*)

5. As seen by reference to Figs. 1, 2 and 3 the plant at the time these tests were made consisted of the producer proper, or generator, an economizer, a scrubber, a centrifugal tar extractor, a purifier and a gas holder. At the present time the plant consists of two independent producers, both discharging into the same scrubber. These producers are worked independently, one being charged while the other is in operation.

6. By this arrangement no time is lost in changing from one coal to another at the end of any test, as a simple manipulation of valves brings the second producer into operation when the first is cut out.

7. Owing to the fact that statements have been published to the effect that no centrifugal tar extractor has proven successful, it gives the writer pleasure to call attention to the fact that the centrifugal extractor used in connection with this plant has proved to be very efficient when properly handled.

8. The details of this latter piece of apparatus are carefully guarded by the manufacturers of the producer, and consequently no detailed drawing is presented. This extractor resembles in outward appearance a centrifugal pump. As the gas is sent through the extractor the tar passes down through the tar drips to a water-sealed pit from which it is easily removed. A liberal supply of water is used during this part of the process of puri-

* This report, known as Professional Paper No. 48, was issued in March, 1906, and may be had upon application to some member of Congress, or to the Director of the Geological Survey.

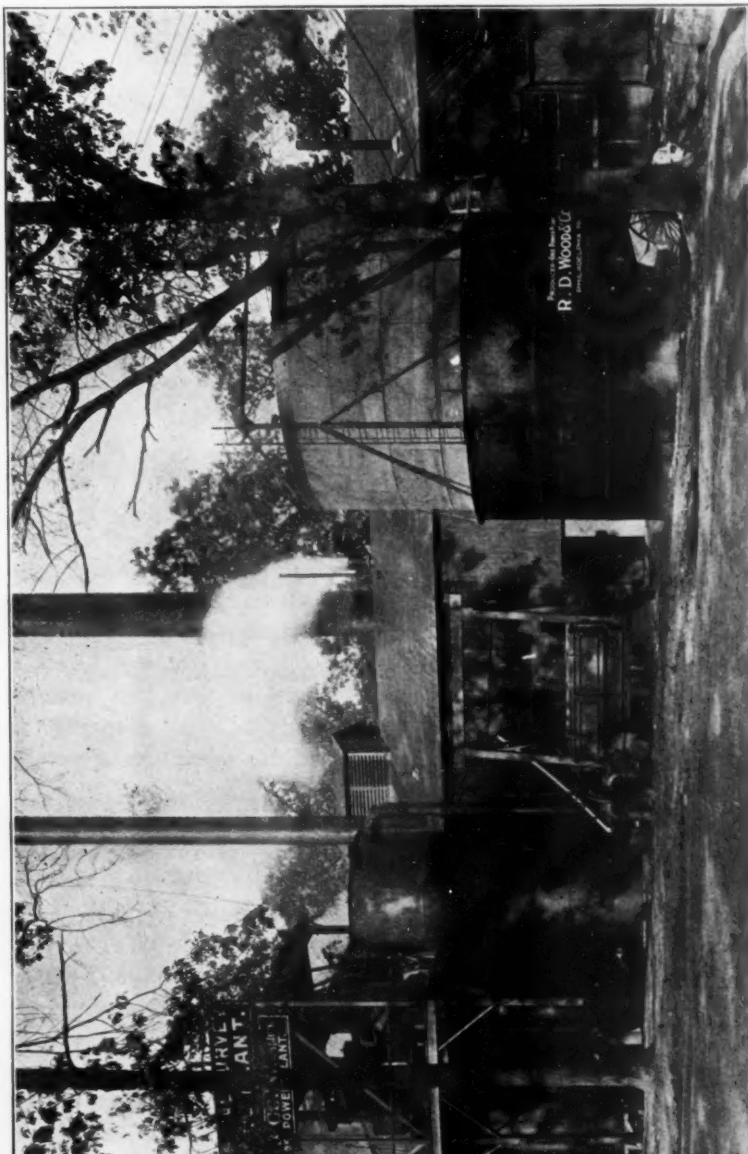


Fig. 1.

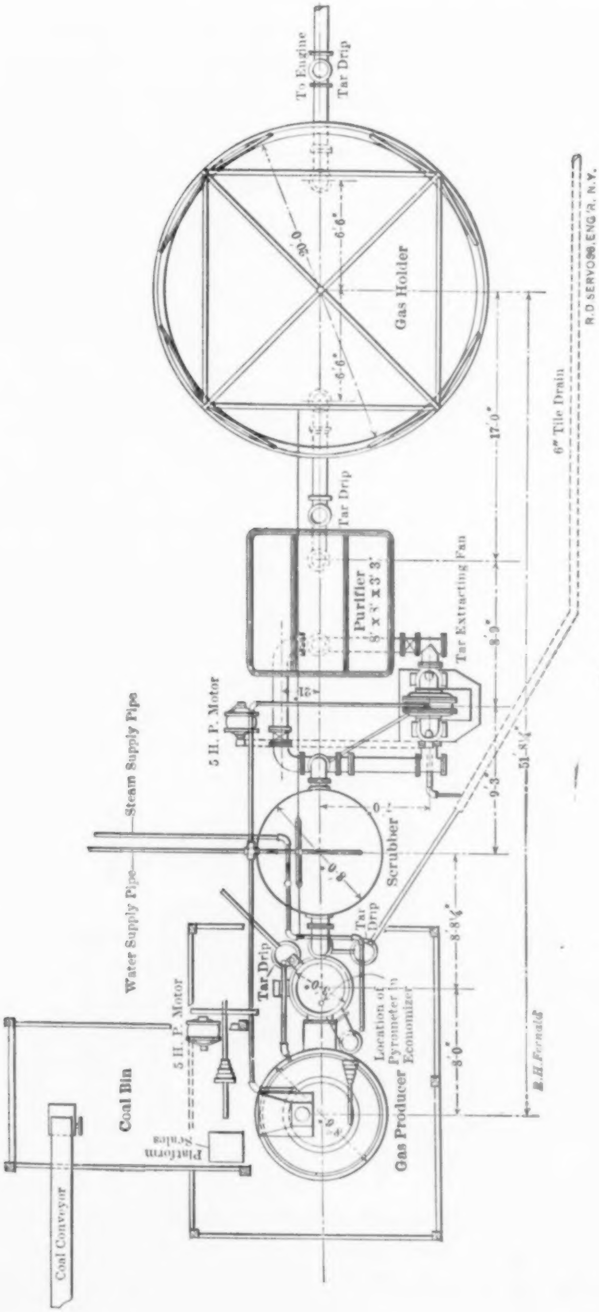

To Engine
Tar Drip
Gas Holder
30'-0"
6'-6"
6'-6"
Tar Drip
Purifier
8'x 4' x 3'3"
Tar Extracting Fan
5 H. P. Motor
Steam Supply Pipe
Water Supply Pipe
Scrubber
Tar Drip
Tar Drip
Location of Pyrometer in Economizer
5 H. P. Motor
Coal Bin
Platform Scales
Gas Producer
Coal Conveyor
6" Tile Drain
17'-0"
8'-9"
9'-3"
8'-8¼"
8'-0"
51'-8¼"
R.H.Fernald
R.D SERVOSS, ENG'R, N.Y.

FIG. 2.

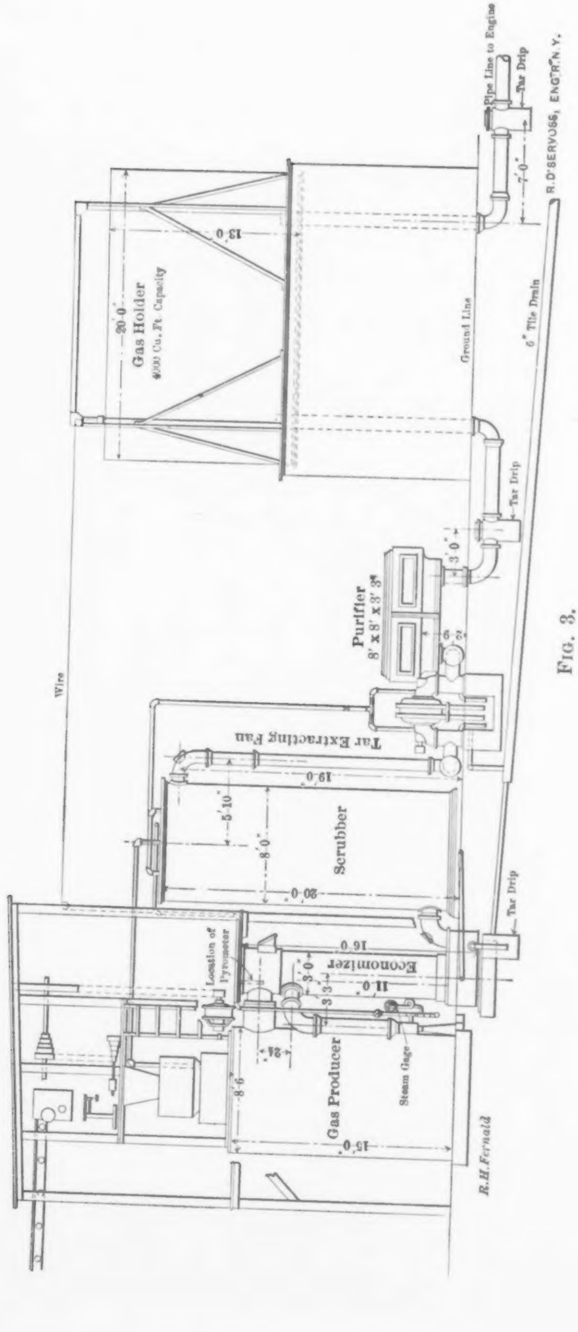

Pipe Line to Engine
Tar Drip
R. D'SERVOSS, ENG'R. N. Y.
7'-0"
Gas Holder
20'-0"
13'-0"
Ground Line
6" Tile Drain
Tar Drip
3'-0"
Purifier
8' x 8' x 3' 3"
Wire
Tar Extracting Fan
19'-0"
5'-10"
8'-0"
Scrubber
20'-0"
Tar Drip
16'-0"
Economizer
Location of Pyrometer
3'-0"
11'-0"
3'-3"
24"
8'-6"
Gas Producer
Steam Gage
15'-0"
R.H.Fernald

FIG. 3.

fication. The speed of rotation of the "fan" in the tar extractor is of vital importance, and any deviation from the proper speed (1,500 to 1,600 revolutions per minute in case of the plant under consideration) seriously affects the successful working of the extractor—particularly if the variation be a reduction of speed. The "fan" in the extractor is driven by a ten-horse-power motor.

9. After going through a meter of 30,000 cubic feet per hour capacity, the gas is delivered to a three-cylinder vertical Westinghouse gas engine. On the pipe line leading from the meter to the engine is a special gas regulator installed by the Westinghouse Co.

10. The cylinders of the engine are 19 inches in diameter and 22 inches in stroke. The engine makes 200 revolutions per minute and is rated at 235 brake horse-power on producer gas.

11. It is belted to a six-pole, 175 kilowatt Westinghouse direct-current generator.

12. The load on the generator is controlled by and the energy developed dissipated through a water rheostat especially constructed for the purpose.

Log and Report Forms Used.

13. In arranging for the tests of the gas producer division of the testing plant, no forms for log or report blanks could be found. It was, therefore, necessary to prepare such forms as would best serve the purpose at the time. These blanks, with slight modifications warranted by the experience of the past few months, are presented in detail.

14. At the time these forms were made out by the writer (September, 1904) few, if any, data could be found relating to the methods of conducting producer gas tests.

15. It is hoped that these forms will be carefully examined and criticized with an idea of securing "standard forms" for reporting producer gas tests.

16. In examining these blanks it should be borne in mind that these tests are producer gas tests from a large series of coals—i. e., the tests are for the purpose of determining the relative values of different bituminous coals and lignites as producer gas developers and they are not "Gas Producer" tests. For this reason it has been the constant aim of the writer to keep to what may be called the "simple problem" and not to attempt the many chemical and thermodynamic problems that present themselves almost daily, interesting and valuable as these might prove.

Form A. Sheet No.

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date,

LOG OF PRODUCER-GAS TEST. No.

Recorded by

Producer made by

Diameter (inside) feet. Height feet.

Rated capacity horsepower.

Coal: General No. Special No.

Car initials and No.

Time.	Coal consumed by producer.	Temperatures, deg. F.					Auxiliary motors.		Pressures.					Water meter.
		Gas.			Air.									
		Leaving producer.	Leaving economizer.	Entering tank.	Entering economizer.	Leaving economizer.	Wattmeter reading.	Amperes.	Air entering producer. (In. mercury.)	Gas leaving producer. (Inches water.)				

Height of gas holder at

Remarks:

17290b1—5-08

Form B. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF FUEL BED CONDITIONS. Test No. ....

Recorded by ....

Producer made by ....

Diameter (inside) .... feet. Height .... feet.

Rated capacity .... horsepower.

Coal: General No. .... Special No. ....

Car initials and No. ....

Height of fuel bed { start .... finish ....

Height of ash bed { start .... finish ....

Character of fire at start ....

Character of ash at start ....

Character of fire at finish ....

Character of ash at finish ....

Remarks: ....

17867b800-3-06

Form C. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF GAS PRODUCER OPERATOR. Test No. ....

Recorded by ....

Producer made by ....

Diameter (inside) .... feet. Height .... feet.

Rated capacity .... horsepower.

Coal: General No. .... Special No. ....

Car initials and No. ....

17898b500-5-06

Form D. Sheet No.

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date,

LOG OF GAS ENGINE TEST. No.

Recorded by

Engine made by

Type of engine

Rated capacity brake horsepower.

Diameter of cylinders inches. Length of stroke inches.

Producer-gas test No.

Coal: General No. Special No.

Car initials and No.

Time.	Speed counters.		Hand counter.	Load output.		Temperature, deg. F.			Pressures.			Gas meter.
						Jacket water.		Entering gas.	Manometers at gas meter. (Inches water.)		Barometer. (Inches mercury.)	
	1.	2.		Volts.	Amperes.	Entering.	Leaving.		1.	2.		

REMARKS:

17399b1m3-09

Form E. Sheet No.

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date,

REPORT OF GAS ENGINE OPERATOR. **Test No.**

Recorded by

Engine made by

Type of engine

Rated capacity brake horsepower.

Diameter cylinders inches. Length of stroke inches.

Producer-Gas Test No.

Coal: General No. Special No.

Car initials and No.

17400b/600-8-08

Form F. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF GAS ENGINE INDICATOR CARDS.

Recorded by ....

Engine made by ....

Type of engine ....

Rated capacity .... brake horsepower.

Diameter of cylinders .... inches. Length of stroke .... inches.

Producer-gas test No. ....

Coal: General No. .... Special No. ....

Car initials and No. ....

Time.	Cylinder.	Card No.	Revolutions per minute.	Scale of spring.	Pressures.				Indicated horsepower.	Total indicated horsepower.
					Comp.	Max.	Rel.	M. E. P.		

REMARKS: ....

10401—09—4—06

Form G. Sheet No. ____

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ____

REPORT OF GAS ENGINE TEST No.

Recorded by ____

Engine made by ____
Type of engine ____
Rated capacity ____ brake horsepower.
Diameter cylinders ____ inches. Length of stroke ____ inches.

Producer-Gas Test No. ____
Coal: General No. ____ Special No. ____
Car initials and No. ____

Duration of test, in hours ____
Revolutions per minute (mean) ____
Explosions per minute (mean) ____
Cubic feet gas per hour, by meter ____
Cubic feet standard gas per hour, (i. e. 62° F., 14.7 lbs. pres.) ____

Pressure at end of compression	1st cylinder	
	2d "	
	3d "	
Maximum pressure	1st "	
	2d "	
	3d "	
Pressure at release	1st "	
	2d "	
	3d "	
Mean effective pressure	1st "	
	2d "	
	3d "	
Indicated horsepower	1st "	
	2d "	
	3d "	

Total indicated horsepower ____
Horsepower delivered (electrical horsepower) ____
Mechanical efficiency (engine and generator combined) ____
Gas horsepower ____
British thermal units equivalent to indicated horsepower ____
British thermal units equivalent to brake horsepower ____
British thermal units equivalent to electrical horsepower ____
British thermal units equivalent to gas horsepower ____
Thermal efficiency, based on indicated horsepower and gas horsepower ____
Thermal efficiency, based on brake horsepower and gas horsepower ____
Thermal efficiency, based on electrical horsepower and gas horsepower ____
Cubic feet standard gas per hour per indicated horsepower ____
Cubic feet standard gas per hour per brake horsepower ____
Cubic feet standard gas per hour per electrical horsepower ____

17493b—3-06

Form 12.   Sheet No.

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date,

LOG OF CALORIMETER TEST.   No.

Recorded by

Producer-gas test No.

Coal: General No.   Special No.

Car initials and No.

Time.	Temperatures.			Cub. cm. water.	Cub. ft. of gas.	Pressure.	British thermal units.	
	Gas.	Water.				Gas, inches water.	As read.	Standardized.
		Inlet.	Outlet.					

Average barometer

Remarks:

[illegible]

Form I. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

LOG OF GAS ANALYSIS.

No. ....

Recorded by ....

Producer gas test No. ....

Coal: General No. .... Special No. ....

Car initials and No. ....

(Per cent by volume.)

Time.										
H_2S										
CO_2										
O_2										
C_2H_4										
CO										
H_2										
CH_4										
N										
TOTAL										
Calculated British thermal units										

REMARKS ....

[illegible]

Form J. Sheet No. ______

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ______

Report of ______

Recorded by ______

Producer-gas test No. ______

Coal: General No. ______ Special No. ______

Car initials and No. ______

17468b1m-8-06

Form K. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF PRODUCER-GAS TEST. No. ....

Recorded by ....

Producer made by ....

Diameter, ... feet. Height, ... feet.

Rated capacity, ... horsepower.

Coal: General No. .... Special No. ....

Car initials and No. ....

1. Duration of test in hours, ...

AVERAGE TEMPERATURE °F

2. Gas leaving producer, ...
3. Gas leaving economizer, ...
4. Gas entering holder, ...
5. Air entering economizer, ...
6. Air leaving economizer, ...

OUTSIDE POWER CHARGED AGAINST PRODUCER PLANT.

7. Total steam used by producer, pounds, ...
8. Steam used by producer per hour, pounds, ...
9. Equivalent in pounds of coal per hour, ...
10. Equivalent in pounds of dry coal per hour, ...
11. Equivalent in pounds of combustible per hour, ...
12. Average horsepower required to drive auxiliary machinery, ...
13. Total water used in scrubber and tar extractor, cubic feet, ...
14. Cubic feet of water per hour per horsepower of producer plant, ...
15. Cubic feet of water per 1,000 cubic feet of gas produced, ...

COAL CONSUMED IN PRODUCER.

16. Total coal consumed, pounds, ...
17. Moisture in coal, per cent, ...
18. Total dry coal consumed, pounds, ...
19. Refuse from coal, per cent, ...
20. Total refuse from coal, pounds, ...
21. Total combustible consumed, pounds, ...

COAL PER HOUR.

22. Coal consumed in producer, pounds, ...
23. Dry coal consumed in producer, pounds, ...
24. Combustible consumed in producer, pounds, ...
25. Equivalent coal used by producer plant, pounds, ...
26. Equivalent dry coal used by producer plant, pounds, ...
27. Equivalent combustible used by producer plant, pounds, ...

Form L. Sheet No. ......

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF PRODUCER-GAS TEST (Continued). No. ....

Recorded by ....

COAL CONSUMED PER SQUARE FOOT OF FUEL BED PER HOUR.

28. Coal as fired, ...
29. Dry coal, ...
30. Combustible, ...

BRITISH THERMAL UNITS FROM COAL.

31. Per pound of coal as fired, ...
32. Per pound of dry coal, ...
33. Per pound of combustible, ...
34. From coal as fired, per hour, ...
35. From dry coal, per hour, ...
36. From combustible, per hour, ...

GAS PRODUCED, CUBIC FEET.
(Gas at 62° F. and 14.7 pounds pressure.)

37. Total, ...
38. Per hour, ...
39. Per pound coal consumed in producer, ...
40. Per pound dry coal consumed in producer, ...
41. Per pound combustible consumed in producer, ...
42. Per pound equivalent coal used by producer plant, ...
43. Per pound equivalent dry coal used by producer plant, ...
44. Per pound equivalent combustible used by producer plant, ...

BRITISH THERMAL UNITS FROM STANDARD GAS.

45. Per cubic foot, ...
46. Per pound dry coal burned in producer, ...
47. Per hour per brake horsepower, ...

AVERAGE HORSEPOWER DEVELOPED.

48. Electrical horsepower available for outside purposes, ...
49. Electrical horsepower developed at switch board, ...
50. Brake horsepower available for outside purposes, ...
51. Brake horsepower developed at engine, ...

EFFICIENCIES.

52. Of conversion and cleaning gas, ...
53. Of producer plant, ...

Form M. Sheet No. ....

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, ...

REPORT OF PRODUCER-GAS TEST (Continued). No. ....

Recorded by ....

COAL PER HORSEPOWER PER HOUR.

	Coal as fired.	Dry coal.	Combustible.
54. Pounds consumed in producer per electrical horsepower available for outside purposes			
55. Pounds consumed in producer per electrical horsepower developed at switch board			
56. Pounds consumed in producer per brake horsepower available for outside purposes			
57. Pounds consumed in producer per brake horsepower developed at engine			
58. Equivalent pounds used by producer plant per electrical horsepower available for outside purposes			
59. Equivalent pounds used by producer plant per electrical horsepower developed at switch board			
60. Equivalent pounds used by producer plant per brake horsepower available for outside purposes			
61. Equivalent pounds used by producer plant per brake horsepower developed at engine			

AVERAGE COMPOSITION OF COAL AND GAS.

62. Coal.

	Per cent.
Moisture	
Volatile matter	
Fixed carbon	
Ash	
Sulphur	

63. Gas by volume.

	Per cent.
Hydrogen disulphide (H_2S)	
Carbon dioxide (CO_2)	
Oxygen (O_2)	
Ethylene (C_2H_4)	
Carbon monoxide (CO)	
Hydrogen (H_2)	
Methane (CH_4)	
Nitrogen (N_2)	

Conditions of Conducting Tests.

17. The tests were begun on a basis of a total of fifty hours for each test. The plant was operated ten hours a day and then fires were banked for the night, the records being continued the next morning. This permitted one test a week only. With the small crew at command it seemed to be the best possible arrangement and was continued for the first two tests. It was then thought desirable to secure double the number of tests, and the schedule was arranged to conduct two tests per week, each of thirty consecutive hours, allowing sufficient time between tests to make the necessary change of fuel and to enable the fuel bed in the producer to be brought to a proper working condition.

18. As it was desired to test as many coals as possible during the few weeks remaining before the close of the Exposition, the highest possible economy was made a secondary consideration, and for a part of the time the plant was run with a leaky hopper and other unfortunate conditions, which naturally impaired its efficiency.

19. In comparing the results it should be borne in mind that in these preliminary tests the object has been to demonstrate the possibility of using these coals in a producer, and not to show how efficiently they could be burned. Although the results in many cases have been highly satisfactory, there is no question that in a second series of tests upon the same coals, made with the idea of showing the greatest economy, the amount of coal per horse-power per hour will, in the majority of cases, be much less.

20. During tests Nos. 5 to 14, inclusive, the hopper of the gas-producer leaked, and considerable gas was wasted, thus vitiating to a small but undetermined extent the efficient results that might otherwise be shown for the coals tested during that period. But at the time of making these tests it was not practicable to stop the operations of the plant for repairs; and the main purpose of the preliminary tests being to determine whether the coals were suitable for producer gas purposes, it was decided to proceed, in spite of the leak in the hopper, and to repeat later, under more favorable conditions, the tests for relative efficiency.

21. Immediately after the close of the Exposition, it having been decided to continue the tests for some weeks longer, the plant was shut down in order to repair the leaking hopper and to prepare for cold weather. Operations were resumed on December 12th, and

continued to December 22d, when a holiday recess was taken. After the recess the tests were continued through the month of January. In all twenty-four producer gas tests were made during the period from the first of October to the first of February.

22. In beginning the new series of tests—May, 1905—a schedule was adopted involving two 60-hour tests per week. This was done to reduce possible errors in determining the amount of coal burned in the producer. The first 8 to 12 hours are now used for getting the fuel bed into a uniform and efficient condition. During this preliminary period records are taken as in the regular tests, but the official test includes only the last 48 or 50 hours of the run, during which time conditions are maintained as uniform as possible.

23. It will be noted that many of the tests reported in this paper were of a few hours' duration only. This was due in many cases to the lack of reliability of the operation of the gas engine, but since the present series of tests began (May, 1905) no difficulty has been experienced in starting the engine at 8 A. M. Monday and continuing day and night without a stop until 8 A. M. Saturday. It should also be noted that two different coals are tested during this period and that the change of gases is made at 8 P. M. Wednesday without stopping the engine.

24. In the following forms, *K, L,* and *M,* will be found the itemized results of one test, viz. No. 15. A test of "Colorado No. 1"—a black lignite which clinkered in the producer, in spite of frequent poking, but the clinkers were not large and could be broken from the top of the producer. The gas was of good, uniform quality, and there is no doubt that the fuel can be used to advantage in producers. It yielded 60 gallons of yellow "lignite tar."

Form K. Sheet No. ......

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, Dec. 13 and 14, 1904.

REPORT OF PRODUCER-GAS TEST. No. 15.

Recorded by W. C. Weidmann, Kurt Toensfeldt, R. E. Feshak.

Producer made by R. D. Wood & Co.

Diameter, (inside) 7 feet. Height, 15 feet.

Rated capacity, 250 horsepower.

Coal: General No. Colorado No. 1, Special No. C. P. 15,

Car initials and No. C. B. & Q. 81529.

1. Duration of test in hours,	30.
AVERAGE TEMPERATURE °F.	
2. Gas leaving producer,	650.
3. Gas leaving economizer,	
4. Gas entering holder,	
5. Air entering economizer,	
6. Air leaving economizer,	
OUTSIDE POWER CHARGED AGAINST PRODUCER PLANT.	
7. Total steam used by producer, pounds,	9,240.
8. Steam used by producer per hour, pounds,	308.
9. Equivalent in pounds of coal per hour,	64.0
10. Equivalent in pounds of dry coal per hour,	51.0
11. Equivalent in pounds of combustible per hour,	47.0
12. Average horsepower required to drive auxiliary machinery,	13.8
13. Total water used in scrubber and tar extractor, cubic feet,	
14. Cubic feet of water per hour per horsepower of producer plant,	
15. Cubic feet of water per 1,000 cubic feet of gas produced,	
COAL CONSUMED IN PRODUCER.	
16. Total coal consumed, pounds,	10,933.
17. Moisture in coal, per cent,	20.24
18. Total dry coal consumed, pounds,	8,720.
19. Refuse from coal, per cent,	7.34
20. Total refuse from coal, pounds,	640.
21. Total combustible consumed, pounds,	8,080.
COAL PER HOUR.	
22. Coal consumed in producer, pounds,	364.4
23. Dry coal consumed in producer, pounds,	290.7
24. Combustible consumed in producer, pounds,	269.3
25. Equivalent coal used by producer plant, pounds,	428.4
26. Equivalent dry coal used by producer plant, pounds,	341.7
27. Equivalent combustible used by producer plant, pounds,	316.6

Form L. Sheet No. ......

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, Dec. 13 and 14, 1904.

REPORT OF PRODUCER-GAS TEST (Continued). No. 15.

Recorded by W. C. Weidmann, Kurt Toensfeldt, R. E. Peshak.

COAL CONSUMED PER SQUARE FOOT OF FUEL BED PER HOUR.

28. Coal as fired,	9.48
29. Dry coal,	7.56
30. Combustible,	7.00

BRITISH THERMAL UNITS FROM COAL.

31. Per pound of coal as fired,	9,767.
32. Per pound of dry coal,	12,245.
33. Per pound of combustible,	13,210.
34. From coal as fired, per hour,	3,560,000.
35. From dry coal, per hour,	3,560,000.
36. From combustible, per hour,	3,560,000.

GAS PRODUCED, CUBIC FEET.
(Gas at 62° F. and 14.7 pounds pressure.)

37. Total,	460,300.
38. Per hour,	15,343.
39. Per pound coal consumed in producer,	42.1
40. Per pound dry coal consumed in producer,	52.8
41. Per pound combustible consumed in producer,	57.0
42. Per pound equivalent coal used by producer plant,	35.8
43. Per pound equivalent dry coal used by producer plant,	44.9
44. Per pound equivalent combustible used by producer plant,	48.5

BRITISH THERMAL UNITS FROM STANDARD GAS.

45. Per cubic foot,	149.0
46. Per pound dry coal burned in producer,	7,860.
47. Per hour per brake horsepower,	9,700.

AVERAGE HORSEPOWER DEVELOPED.

48. Electrical horsepower available for outside purposes,	186.4
49. Electrical horsepower developed at switch board,	200.2
50. Brake horsepower available for outside purposes,	219.3
51. Brake horsepower developed at engine,	235.4

EFFICIENCIES.

52. Of conversion and cleaning gas,	.642
53. Of producer plant,	.543

Form M. Sheet No.........

U. S. GEOLOGICAL SURVEY COAL-TESTING PLANT.

GAS PRODUCER DIVISION.

Date, Dec. 13 and 14, 1904.

REPORT OF PRODUCER-GAS TEST (Continued). No. 15.

Recorded by W. C. Weidmann, Kurt Toensfeldt, R. E. Peshak.

COAL PER HORSEPOWER PER HOUR.

	Coal as fired.	Dry coal.	Combustible.
54. Pounds consumed in producer per electrical horsepower available for outside purposes	1.95	1.56	1.45
55. Pounds consumed in producer per electrical horsepower developed at switch board	1.82	1.45	1.35
56. Pounds consumed in producer per brake horsepower available for outside purposes	1.66	1.32	1.23
57. Pounds consumed in producer per brake horsepower developed at engine	1.55	1.23	1.14
58. Equivalent pounds used by producer plant per electrical horsepower available for outside purposes	2.30	1.83	1.70
59. Equivalent pounds used by producer plant per electrical horsepower developed at switch board	2.14	1.71	1.58
60. Equivalent pounds used by producer plant per brake horsepower available for outside purposes	1.95	1.56	1.44
61. Equivalent pounds used by producer plant per brake horsepower developed at engine	1.82	1.45	1.34

AVERAGE COMPOSITION OF COAL AND GAS.

62. Coal.

	Per cent.
Moisture	20.24
Volatile matter	32.26
Fixed carbon	41.65
Ash	5.85
	100.00
Sulphur	.60

63. Gas by volume.

	Per cent.
Hydrogen disulphide (H_2S)	
Carbon dioxide (CO_2)	10.11
Oxygen (O_2)	.55
Ethylene (C_2H_4)	
Carbon monoxide (CO)	17.38
Hydrogen (H_2)	11.05
Methane (CH_4)	5.00
Nitrogen (N_2)	55.91
	100.00

Summary of Results.

25. In view of the fact that the United States Geological Survey will issue in a few weeks a complete * report, in which the writer gives a detailed explanation of each item mentioned above, it is deemed wise to present results only in this paper.

26. The table given herewith shows in a condensed form the results obtained in the first twenty-four producer gas tests.

27. The chart, Fig. 4, which follows the table presents in a compact manner the most important items. As will be seen at a glance the coals are arranged in the order of their economic value as determined by the tests already made. The order of precedence may be slightly modified in the light of future tests, conducted under improved conditions, although it is doubtful if it is seriously shaken.

28. Attention is called to the fact that in these tests all hydrocarbons have been determined and figured as methane. Small amounts of ethylene, probably occurring in the gas, have in this way been figured as methane, slightly decreasing the calorific values of the gas calculated. It was deemed inadvisable to attempt the determination of ethylene by means of fuming sulphuric acid, owing to the fact that the laboratory provided for these determinations was so small that acid could not satisfactorily be used. Bromine was tried for the ethylene absorption, but found to be of no value owing to the small amounts of this gas present. Since the new chemical laboratory was installed in April no such cramped conditions have existed and ethylene has been separately determined.

29. Although some of the coals used in these preliminary tests made excellent records, yet the results obtained since beginning the second series of tests in May, 1905, have been far more satisfactory, and it is gratifying to report that official records have been made as low as 0.95 pounds of dry coal per hour burned in the producer per electrical horse-power developed at the switchboard; or 0.80 pounds of dry coal per hour burned in the producer per brake horse-power per hour, on the basis of an efficiency of 85 per cent. for generator and belt.

Attention is also called to the fact that two tests upon the same

* This report, known as Professional Paper No. 48, was issued in March, 1906, and may be had upon application to some member of Congress, or to the Director of the Geological Survey.

BRIEF SUMMARY OF PRODUCER GAS TESTS OF TWENTY-FOUR BITUMINOUS COALS AND LIGNITES

Tests.			Name of sample.	Coal per Hour, Pounds.						British Thermal Units.		
				Consumed in producer.			Equivalent used by producer plant.			Per pound.		
No. of test.	Date of test.	Duration in hours.		Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.
1	2	3	4	5	6	7	8	9	10	11	12	13
2	Oct. 10–14	43.00	Alabama No. 2	310.5	299.0	280.0	341.4	328.7	306.8	12,865	13,365	14,820
15	Dec. 13–14	30.00	Colorado No. 1	364.4	290.7	269.3	428.4	341.7	316.6	9,767	12,245	13,210
a 6	Oct. 31–Nov. 1	30.00	Illinois No. 3	350.0	323.3	289.3	386.0	356.7	319.2	12,046	13,041	14,560
a 9	Nov. 10–11	30.00	Illinois No. 4	350.0	306.3	274.1	398.2	348.5	311.9	11,237	12,834	14,344
a 14	Dec. 2–3	29.67	Indiana No. 1	394.5	349.3	309.5	434.6	384.8	341.0	11,534	13,037	14,720
a 13	Nov. 28–29	7.00	Indiana No. 2	300.0	274.0	244.8	338.0	312.0	278.8	11,822	12,953	14,500
1	Oct. 3–6	31.00	Indian Territory No. 1	361.0	344.0	312.0	392.7	374.0	339.3	12,787	13,455	14,800
20	Jan. 23–24	22.67	Indian Territory No. 4	278.0	253.2	207.8	312.5	284.6	233.6	10,364	11,392	13,890
22	Jan. 30–31	13.33	Iowa No. 2	362.5	302.5	227.5	408.4	340.7	256.2	8,735	10,489	13,950
24	Feb. 6–7	13.00	Kansas No. 5	307.8	294.3	259.8	338.4	323.6	285.7	12,836	13,421	15,200
a 12	Nov. 25–26	30.00	Kentucky No. 3	370.0	343.3	310.0	410.8	381.2	344.2	12,283	13,226	14,650
a 7	Nov. 3	4.33	Missouri No. 2	346.5	306.0	255.0	384.5	339.6	283.0	10,505	11,882	14,280
a 5	Oct. 24–25	22.33	Montana No. 1	456.5	404.5	355.8	506.8	449.1	395.0	10,575	11,934	13,580
a 10	Nov. 14–15	30.00	North Dakota No. 2	460.0	278.0	249.0	510.0	308.0	275.8	6,802	11,255	12,600
a 11	Nov. 21–22	21.67	Texas No. 1	590.0	393.0	332.0	660.0	439.5	371.3	7,267	10,928	12,945
18	Dec. 21–22	19.33	Texas No. 2	468.0	310.3	276.2	519.5	344.4	306.6	7,348	11,086	12,450
3	Oct. 17–18	24.00	West Virginia No. 1	287.5	283.0	265.5	320.6	315.6	296.1	14,166	14,396	15,350
4	Oct. 20	9.00	West Virginia No. 4	233.0	229.0	208.0	262.8	258.2	234.5	13,918	14,202	15 600
23	Feb. 2–3	22.67	West Virginia No. 7	264.9	256.9	239.1	299.2	290.2	270.1	14,283	14,720	15,800
19	Jan. 20–21	21 00	West Virginia No. 8	328.6	320.1	301.1	364.7	355.1	334.1	14,168	14,558	15,470
21	'an. 27–28	24 00	West Virginia No. 9	250.0	244.5	227.0	284.8	278.5	258.6	14,224	14,548	15,650
17	Dec. 19	6.33	West Virginia No. 9	300.0	292.0	274.9	328.9	320.1	301.4	14,195	14,580	15,500
a 8	Nov. 7–8	30.00	West Virginia No. 12	270.0	266.1	248.7	304.9	300.5	280.9	14,614	14,825	15,860
[illegible]	Dec. 16–17	30 00	Wyoming No. 2	403.3	365.3	281.6	459.8	416.5	321.1	9,650	10,656	13,820

a Gas producer hopper leaked during these tests.

BRIEF SUMMARY OF PROCURER GAS TESTS OF TWENTY-FOUR BITUMINOUS COALS AND LIGNITES.

Tests.	British Thermal Units.			Cubic Feet Standard Gas Produced. (62° F., 14.7 pounds pressure.)							Economic Results. (Pounds of coal consumed in producer per h.-p. per hour.)		
					Per pound consumed in producer.			Per pound equivalent used by producer plant.			Per electrical horse-power available for outside purposes.		
No. of test.	Per cubic foot of standard gas.	From standard gas per pound of dry coal consumed in producer.	From standard gas per hour per brake horse-power.	Per hour.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.
1	14	15	16	17	18	19	20	21	22	23	24	25	26
2	149.2	9,000	11,420	18,050	58.1	60.4	64.5	52.9	55.0	58.9	1.61	1.55	1.45
15	149.0	7,860	9,700	15,343	42.1	52.8	57.0	35.8	44.9	48.5	1.95	1.56	1.45
a 6	154.8	8,330	11,460	17,412	49.8	53.9	60.2	45.1	48.8	54.5	1.82	1.68	1.50
a 9	151.5	8,840	11,620	17,881	51.1	58.4	65.3	44.9	51.4	57.4	1.85	1.62	1.45
a 14	153.7	7,730	11,480	17,560	44.5	50.3	56.7	40.4	45.6	51.5	2.10	1.86	1.64
a 13	159.3	10,140	11,750	17,450	58.2	63.6	71.3	51.6	55.9	62.6	1.57	1.43	1.28
1	159.2	8,620	12,350	18,613	51.6	54.1	59.4	47.4	49.9	54.6	1.84	1.76	1.59
20	161.1	9,980	10,750	15,680	56.4	61.9	75.5	50.2	55.1	67.1	1.48	1.35	1.10
22	160.2	9,300	12,130	17,570	48.5	58.1	77.2	43.0	51.6	68.5	1.94	1.62	1.22
24	167.2	10,500	13,130	18,490	60.1	62.8	71.2	54.6	57.2	64.8	1.60	1.53	1.35
a 12	155.9	8,610	12,540	18,943	51.2	55.1	61.1	46.2	49.7	55.0	1.95	1.81	1.63
a 7	140.0	8,820	11,560	19,300	55.7	63.0	75.7	50.2	56.8	68.2	1.87	1.65	1.37
a 5	160.8	6,580	11,340	16,540	36.2	40.9	46.5	32.6	36.8	41.9	2.39	2.12	1.86
a 10	188.5	7,830	13,770	11,550	25.2	41.5	46.4	22.7	37.5	41.9	3.67	2.22	1.99
a 11	169.7	7,260	12,230	16,800	28.4	42.7	50.6	25.5	38.2	45.3	3.16	2.10	1.78
18	156.2	8,060	10,570	16,009	34.2	51.6	57.9	30.8	46.4	52.2	2.47	1.64	1.46
3	144.4	9,260	11,130	18,150	63.2	64.1	68.4	56.6	57.5	61.3	1.51	1.49	1.40
4	143.2	11,610	11 320	18,560	79.6	81.2	89.2	70.6	71.9	79.2	1.23	1.21	1.10
23	154.2	13,140	14 380	21,850	82.5	85.1	91.4	73.0	75.3	80.9	1.41	1.37	1.27
19	155.1	9,070	12,340	18,690	56.9	58.4	62.0	51.3	52.6	55.9	1.73	1.68	1.58
21	160.5	11,380	11,860	17,330	69.3	70.9	76.3	60.9	62.2	67.0	1.32	1.29	1.20
17	151.0	8,150	10,060	15,770	52.6	54.0	57.4	48.0	49.3	52.3	1.60	1.56	1.47
a 8	142.5	10,150	11,500	18,957	70.2	71.2	76.2	62.1	63.2	67.5	1.41	1.39	1.30
6	151.0	6,168	9,516	14,923	37.0	40.9	53.0	32.5	35.8	46.5	2.18	1.98	1.52

a Gas producer hopper leaked during these tests.

BRIEF SUMMARY OF PRODUCER GAS TESTS OF TWENTY-FOUR BITUMINOUS COALS AND LIGNITES.

Tests.	Economic Results [continued]. (Pounds of coal consumed in producer per horse-power per hour.)									Economic Results. (Equivalent pounds of coal used by producer plant per horse-power per hour.)					
	Per electrical horse-power developed at switch board.			Per brake* horse-power available for outside purposes.			Per brake* horse-power developed at engine.			Per electrical horse-power available for outside purposes.			Per electrical horse-power developed at switch board.		
No. of test.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.
1	**27**	**28**	**29**	**30**	**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**	**41**
2	1.55	1.49	1.40	1.37	1.32	1.23	1.32	1.27	1.19	1.77	1.71	1.59	1.71	1.64	1.53
15	1.82	1.45	1.35	1.66	1.32	1.23	1.55	1.23	1.14	2.30	1.83	1.70	2.14	1.71	1.58
a 6	1.75	1.62	1.45	1.54	1.43	1.28	1.49	1.38	1.23	2.01	1.85	1.66	1.93	1.79	1.60
a 9	1.76	1.55	1.38	1.57	1.37	1.23	1.50	1.31	1.17	2.11	1.84	1.65	2.01	1.76	1.57
a 14	1.97	1.75	1.55	1.78	1.58	1.40	1.68	1.49	1.32	2.31	2.04	1.81	2.17	1.93	1.71
a 13	1.49	1.36	1.22	1.33	1.22	1.09	1.27	1.16	1.03	1.77	1.63	1.46	1.68	1.55	1.39
1	1.77	1.69	1.53	1.57	1.50	1.36	1.50	1.43	1.30	2.00	1.91	1.73	1.92	1.83	1.66
20	1.39	1.27	1.04	1.26	1.15	.94	1.18	1.08	.89	1.66	1.51	1.24	1.57	1.43	1.17
22	1.84	1.53	1.15	1.65	1.38	1.04	1.56	1.30	.98	2.19	1.83	1.37	2.07	1.73	1.30
24	1.54	1.47	1.30	1.36	1.30	1.15	1.31	1.25	1.10	1.76	1.68	1.49	1.69	1.62	1.43
a 12	1.85	1.71	1.55	1.66	1.54	1.39	1.57	1.46	1.32	2.16	2.01	1.81	2.05	1.91	1.72
a 7	1.74	1.54	1.28	1.59	1.40	1.17	1.48	1.31	1.09	2.07	1.83	1.52	1.94	1.71	1.43
a 5	2.29	2.03	1.78	2.03	1.80	1.58	1.95	1.72	1.52	2.65	2.35	2.07	2.54	2.25	1.98
a 10	3.42	2.07	1.86	3.13	1.89	1.69	2.91	1.76	1.58	4.07	2.46	2.20	3.80	2.29	2.05
a 11	2.98	1.99	1.68	2.68	1.79	1.51	2.54	1.69	1.43	3.53	2.35	1.99	3.34	2.22	1.88
18	2.33	1.54	1.37	2.10	1.39	1.24	1.98	1.31	1.17	2.74	1.82	1.62	2.58	1.71	1.52
3	1.43	1.41	1.33	1.29	1.27	1.19	1.22	1.20	1.13	1.69	1.66	1.56	1.60	1.57	1.48
4	1.17	1.15	1.04	1.05	1.03	.94	.99	.98	.89	1.39	1.36	1.24	1.32	1.29	1.17
23	1.33	1.29	1.20	1.20	1.16	1.08	1.13	1.10	1.02	1.59	1.55	1.44	1.50	1.46	1.36
19	1.64	1.60	1.51	1.47	1.43	1.35	1.40	1.36	1.28	1.92	1.87	1.76	1.82	1.78	1.67
21	1.26	1.23	1.14	1.18	1.10	1.02	1.07	1.04	.97	1.51	1.47	1.37	1.43	1.40	1.30
17	1.49	1.46	1.37	1.36	1.33	1.25	1.27	1.24	1.16	1.76	1.71	1.61	1.64	1.59	1.50
a 8	1.35	1.33	1.24	1.20	1.18	1.11	1.15	1.13	1.06	1.59	1.57	1.47	1.53	1.50	1.40
6	2.00	1.82	1.40	1.86	1.68	1.30	1.70	1.54	1.19	2.49	2.25	1.74	2.28	2.07	1.60

a Gas producer hopper leaked during these tests.

* Based on an assumed efficiency of 85% for generator and belt.

BRIEF SUMMARY OF PRODUCER GAS TESTS OF TWENTY-FOUR BITUMINOUS COALS AND LIGNITES.

Tests.	Economic Results [continued]. (Equivalent pounds of coal used by producer plant per horse-power per hour.)						Average Composition of Gas by Volume, per Cent.						Name of sample.	No. of test.
	Per brake* horse-power available for outside purposes.			Per brake* horse-power developed at engine.										
No. of test.	Coal as fired.	Dry coal.	Combustible.	Coal as fired.	Dry coal.	Combustible.	Carbonic acid, CO_2.	Oxygen, O_2.	Carbonic oxide, CO.	Hydrogen, H_2.	Marsh gas, CH_4.	Nitrogen, N_2.		
1	**42**	**43**	**44**	**45**	**46**	**47**	**48**	**49**	**50**	**51**	**52**	**53**	**4**	**1**
2	1.51	1.45	1.35	1.45	1.40	1.30	8.16	.10	16.65	7.20	5.64	62.24	Alabama No. 2	2
15	1.95	1.56	1.44	1.82	1.45	1.34	10.11	.55	17.38	11.05	5.00	55.90	Colorado No. 1	15
a 6	1.70	1.58	1.41	1.64	1.52	1.36	10.53	.15	15.31	8.35	4.46	61.19	Illinois No. 3	*a* 6
a 9	1.79	1.56	1.40	1.71	1.50	1.34	9.72	.12	15.12	9.98	6.00	59.06	Illinois No. 4	*a* 9
a 14	1.96	1.74	1.54	1.85	1.64	1.45	9.89	.25	14.10	9.56	6.08	60.13	Indiana No. 1	*a* 14
a 13	1.52	1.39	1.24	1.43	1.32	1.18	11.80	.07	11.46	10.60	6.10	59.97	Indiana No. 2	*a* 13
1	1.71	1.62	1.47	1.64	1.56	1.41	8.25	.11	19.39	7.69	4.92	59.65	Indian Territory No. 1	1
20	1.41	1.29	1.06	1.33	1.21	1.00	7.29	.24	17.64	10.43	6.30	58.10	Indian Territory No. 4	20
22	1.86	1.55	1.17	1.76	1.47	1.10	10.06	.17	12.57	9.53	7.67	60.00	Iowa No. 2	22
24	1.50	1.43	1.26	1.44	1.37	1.21	10.27	.13	12.40	9.05	7.42	60.73	Kansas No. 5	24
a 12	1.84	1.71	1.54	1.75	1.62	1.46	10.87	.29	12.45	10.92	6.52	58.95	Kentucky No. 3	*a* 12
a 7	1.76	1.55	1.30	1.65	1.45	1.21	12.07	.20	10.53	7.63	6.33	63.23	Missouri No. 2	*a* 7
a 5	2.26	2.00	1.76	2.16	1.91	1.68	9.04	.36	18.67	8.00	4.84	59.10	Montana No. 1	*a* 5
a 10	3.47	2.09	1.88	3.23	1.95	1.74	8.69	.23	20.90	14.33	4.85	51.02	North Dakota No. 2	*a* 10
a 11	3.00	2.20	1.69	2.83	1.99	1.60	11.10	.22	14.43	10.54	7.48	56.22	Texas No. 1	*a* 11
18	2.33	1.55	1.38	2.20	1.46	1.30	9.60	.20	18.22	9.63	4.81	57.53	Texas No. 2	18
3	1.43	1.41	1.33	1.36	1.34	1.26	10.50	.10	14.34	2.81	5.56	66.69	West Virginia No. 1	3
4	1.18	1.16	1.05	1.12	1.10	1.00	10.16	.24	15.82	11.16	3.74	58.88	West Virginia No. 4	4
23	1.35	1.31	1.22	1.28	1.24	1.15	9.62	.08	12.75	10.31	6.76	60.48	West Virginia No. 7	23
19	1.63	1.59	1.49	1.55	1.51	1.42	10.33	.22	11.93	9.45	6.40	61.67	West Virginia No. 8	19
21	1.28	1.25	1.16	1.21	1.19	1.10	8.90	.33	14.77	9.52	6.65	59.83	West Virginia No. 9	21
17	1.49	1.46	1.37	1.39	1.35	1.27	10.40	.20	13.70	9.55	6.60	59.55	West Virginia No. 9	17
a 8	1.35	1.34	1.25	1.30	1.28	1.20	10.34	.12	14.21	12.98	4.61	57.75	West Virginia No. 12	*a* 8
6	2.11	1.92	1.48	1.94	1.76	1.36	10.21	.59	15.46	10.79	5.52	57.43	Wyoming No. 2	16

a Gas producer hopper leaked during these tests.

* Based on an assumed efficiency of 85% for generator and belt.

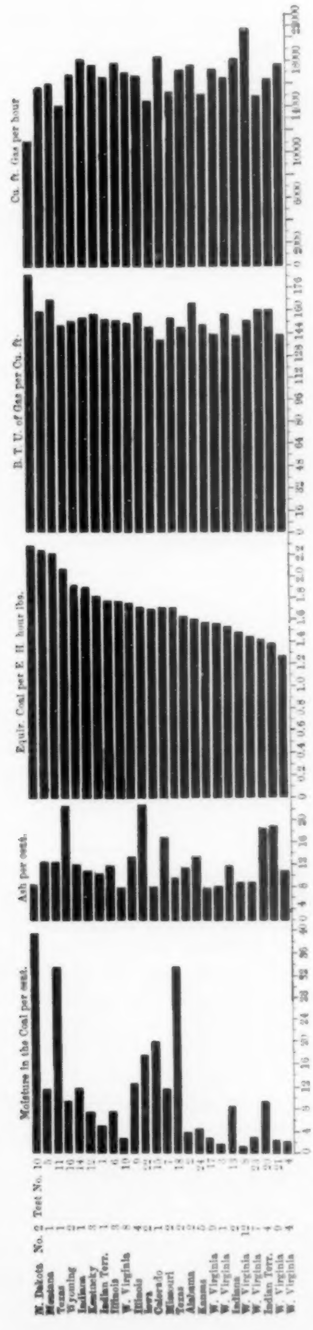

Cu. ft. Gas per hour
B. T. U. of Gas per Cu. ft.
Equiv. Coal per E. H. hour lbs.
Ash per cent.
Moisture in the Coal per cent.
Test No.

FIG. 4.

coal made on different dates and in the two different producers now in operation, at the plant, gave results that checked beyond all expectations.

Comparison of Results Obtained from Steam and Producer Gas Tests.

30. The following quoted passages are from the statements of Prof. L. P. Breckenridge as reported on page 118 of U. S. Geological Survey, Bulletin No. 261:—

"It is to be recollected that the steam generated by the boiler was used in a simple non-condensing engine of the Corliss type, whose "water rate" was 26.3 pounds of steam per hour per horse-power developed; that this engine was belted to the electric generator, and that the mechanical efficiency of this combination of engine and generator was 81 per cent.

"With these figures available it will be an easy matter to calculate the number of pounds of coal which would have been required to produce an electrical horse-power provided a more economical type of steam engine had been used, or if the electrical generator had been directly connected to the engine, with the resulting advantage of a higher mechanical efficiency.

"If, for example, the steam generated had been used by a steam engine capable of generating 1 horse-power with 18 pounds of steam per hour, and if the engine and generator had been direct connected, giving as high a mechanical efficiency as 90 per cent., then the 'Total dry coal per electrical horse-power per hour' would have been reduced from 4.3 pounds, as given in column 13, to very nearly 3 pounds.

"While these figures are frequently and easily attained by steam engines operating in large units, it will be conceded that in plants of from 200 to 250 horse-power they are but seldom reached.

"It should be mentioned that the labor required would be the same for the operation of either the boiler plant or the gas producer plant of the capacity under tests. In either plant two men would be sufficient.

"In considering the possible increase in efficiency of the boiler tests with a compound engine substituted for the simple engine used, the fact should not be overlooked that a corresponding increase in the efficiency of the gas producer tests may be brought about under the most favorable conditions. The gas engine is passing through a transitional period. In the larger sizes the vertical single-acting engine is being replaced by the horizontal double-acting. Other changes and improvements are constantly being made which tend to do for the gas engine what compounding and tripling the expansions have done for the steam engine.

"The gas engine used in the trials recorded is a vertical three-cylinder, single-acting engine with no means of changing the ignition while the engine is running."

31. In the tests at hand, a glance at the "British thermal units per hour per brake horse-power" will show in many instances that the ignition setting was faulty, thus causing an excessive amount of gas to be used. With a variable ignition device there

COMPARATIVE SUMMARY OF THE LEADING RESULTS OF THE COAL TESTS MADE UNDER THE BOILER AND IN THE GAS PRODUCER.

Name of Sample.	Duration of Trial.		Total Dry Coal Consumed per Hour.*		Dry Coal Burned per Sq. Ft. of Grate Surface per Hour.		Water Evaporated from and at 212° F. per Lb. of Dry Coal.	B.T.U. per Pound of Dry Coal Used.		Electrical Horse-power Delivered to Switchboard.		Total Dry Coal per Electrical Horse-power per Hour.*		Ratio of Item 13 to Item 14.
	Steam Plant.	Gas Producer Plant.	Steam Plant.	Gas Producer Plant.	Steam Plant.	Gas Producer Plant.†	Steam Plant.	Steam Plant.	Gas Producer Plant.	Steam Plant.	Gas Producer Plant.	Steam Plant.	Gas Producer Plant.	
	Hours.	Hours.	Pounds.	Pounds.	Pounds.	Pounds.	Pounds.					Pounds.	Pounds.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Alabama No. 2.	10.02	43.00	874	328.7	21.54	7.78	8.55	12555	13365	213.7	200.6	4.08	1.64	2.49
Colorado No. 1..	9.97	30.00	722	341.7	17.80	7.56	7.21	12577	12245	149.1	200.2	4.84	1.71	2.83
Illinois No. 3‡ ..	10.13	30.00	861	356.7	21.23	8.41	8.04	12857	13041	198.1	199.6	4.34	1.79‡	2.42
Illinois No. 4‡ ..	10.02	30.00	938	348.5	23.13	7.96	7.27	12459	12834	195.4	198.4	4.80	1.76‡	2.73
Indiana No. 1‡.	9.93	29.67	908	384.3	22.39	9.08	8.45	13377	13037	220.0	199.9	4.13	1.93‡	2.14
Indiana No. 2‡	10.13	7.00	832	312.0	20.51	7.13	8.02	12452	12953	191.0	201.0	4.35	1.55‡	2.81
Ind. Ter. No. 1.	9.75	31.00	778	374.0	19.17	8.95	8.64	12834	13455	192.3	204.0	4.04	1.83	2.21
Ind. Ter. No. 4.	10.22	22.67	853	284.6	21.04	6.59	7.53	12027	11392	184.0	199.5	4.64	1.43	3.24
Iowa No. 2	9.92	13.33	945	340.7	23.28	7.87	7.05	11497	10489	190.9	197.5	4.95	1.73	2.86
Kansas No. 5 ..	9.90	13.00	837	323.6	20.62	7.65	8.88	13144	13421	212.8	200.2	3.93	1.62	2.43
Kentucky No.3‡	10.07	30.00	882	381.2	21.75	8.92	8.27	13036	13226	208.9	200.5	4.22	1.91‡	2.21
Missouri No. 2‡.	9.98	4.33	1014	339.6	25.00	7.96	7.08	11500	11882	205.6	198.6	4.93	1.71‡	2.88
W. Va. No. 1 ..	9.98	24.00	768	315.6	18.94	7.36	8.95	14198	14396	196.7	200.4	3.90	1.57	2.48
W. Va. No. 4...	10.00	9.00	770	258.2	18.98	5.96	9.65	14002	14202	212.5	199.7	3.62	1.29	2.80
W. Va. No. 7...	10.18	22.67	736	290.2	18.15	6.68	9.85	14305	14720	207.5	199.1	3.55	1.46	2.43
W. Va. No. 8...	9.98	21.00	760	355.1	18.72	8.32	9.62	14436	14558	209.0	199.9	3.63	1.78	2.04
W. Va. No. 9...	10.00	24.00	721	278.5	17.78	6.36	10.09	14616	14548	208.2	199.3	3.46	1.40	2.47
W. Va. No. 12‡.	10.13	30.00	719	300.5	17.68	6.92	9.90	15170	14825	203.6	199.8	3.53	1.50‡	2.35
Wyoming No. 2	9.95	30.00	1075	416.5	26.51	9.50	5.92	10897	10656	182.0	201.2	5.90	2.07	2.85

Average.............. 2.57

* In gas producer plant this includes the coal consumed in the producer and the coal equivalent of the steam used in operating the producer.
† Coal actually consumed in producer only. ‡ Gas producer hopper leaked during these tests.

is no doubt that marked improvement can be shown in the amount of gas used by the engine.

32. The producer itself was subjected to new and varied conditions for which it was not well adapted. In the light of the information which has been obtained during the operation of this plant together with the changes and improvements that will naturally be made in the construction and operation of producer plants for use in connection with bituminous coals and lignites, there is every reason to expect a development in this particular line which will make a marked increase in the efficiency of such installations.

33. A brief consideration of these points will lead at once to the conclusion that the gas engine and steam engine used in these tests compare very favorably, and that any increase in efficiency in the boiler tests that might result from using a compound engine can be offset by the introduction of the more modern type of gas engine and the improved producer plant.

DISCUSSION.

Mr. F. E. Junge.—If I take the liberty as a guest of this distinguished Society, and as a member of the Society of German Engineers of Berlin, of taking part in this discussion, it is not because I find something to criticise in the very interesting report as presented by Professor Fernald, but because my country has for several years past been credited by the American technical press with having done some good investigation and research work in connection with the problems pertaining to this particular branch of engineering, and because I think that it may prove of interest to you to learn of a few results which have lately been recorded to me.

You will have noticed from the tests made by the United States Geological Survey that in the generation of power from low grade fuels, such as peat and lignite, or what we call brown coal, a great quantity of tar is produced as a by-product, which, on account of its low salability in this country, cannot be efficiently utilized. We have recently devised means whereby the tar is burned immediately after its production, and in the generator itself, by having two combustion zones, one at the top and one at the bottom, of the producer chamber, and aspirating air from above as well as from below while the gas is discharged in the middle, between the two spheres of combustion.

We thus have in the upper burning layers the Siemens process, generating gas which is free from heavy hydro-carbons as these are burned in the incandescent zone and containing mainly carbon monoxide and nitrogen, while in the lower bed the Dowson process is used, generating a gas which is mixed and burned together with the Siemens gas from above. In these double producers we have encountered no difficulty with caking, and the gas generated is so free from tar that a single coke scrubber is sufficient for carrying out the entire process of washing and cleaning.

With brown coal bricquettes, having a calorific value of 5100 Cal. per Kg. (9180 British thermal units per pound), we can produce one British horse-power hour on 0.50 kg. (1.1 pound) of lignite. This is rather a satisfactory result with this low-grade fuel. Of course with anthracite coal we come down to a much lower consumption. For instance, in the Gueldner suction gas plant and engine we attain with a coal, having a heat value of 7700 Cal. per Kg. (13,860 British thermal units per pound) one British horse-power hour on 0.334 Kg. (0.73 pounds), that is less than ¾ pound brake horse-power per hour. It may be added that the above figures are not taken from a show test, but were attained under actual working conditions.

Turning to an even more interesting feature of producer practice, I want to say that it is now quite within the range of practicability to dispose of city garbage and sewage by burning it in producers, thus transforming it into useful power. The method consists in admixing to the garbage and sewage in the city pipes a small percentage of powdered lignite, which absorbs the humus particles of the waste, and certain ferro-aluminium and magnesia salts to destroy such molecules as resist absorption. The solid part of the mixture is then allowed to settle down in ponds, is dried afterwards and burned in the producer. With this method we can generate one British horse-power hour from two Kg. (4.4 pounds) of garbage and sewage. A city of fifty thousand inhabitants produces daily about 12,000 Kg. (26,400 pounds) of waste products from which can be generated 6,000 horse-power hours, or 1,460,000 kilowatt hours a year. This process is apparently one of the cleanest methods of disposing of city garbage, and I wish to recommend it to the attention of American engineers.

I also want to say a few words about the cleaning of producer gas, which is so indissolubly connected with its generation and so indispensable for its successful utilization in gas engines that it must

logically be discussed in a paper on producer practice. I have had the opportunity of visiting and testing quite a number of gas power plants in this country, and among others have studied the plant of the Lackawanna Steel Works, in Buffalo, in which 40,000 horse-power are produced from generators and the waste gases of the blast furnaces. I shall refrain from any criticism on the installation of this plant, but I believe it to be generally known that the scheme has not given complete satisfaction. In talking over this matter with some gentlemen from the Steel Trust, and discussing their reluctant attitude toward the general subject of the utilization of waste gases from coke ovens and blast furnaces, I found that they almost unanimously attributed the failure of such plants to the deficiency of the gas engine to come up to the fundamental requirement of reliability. This notion I found also prevailing among the more conservative set of engineers of this country.

With the experience of the German industry at the back of me, which has absolutely passed out of the transitory state of costly experiment and has reached a condition of profitable balance, I want to emphasize at this place what I try to prove by contributions to the technical papers of this country, i.e., that the large gas engine, and particularly such types as built by the Allis-Chalmers Company, of Milwaukee, and by the De La Vergne Company, of this city, with which I am most familiar, have arrived at a standardization of form and reliability of operation which rivals in excellence that of the highest class of Corliss and other steam engine makes. I have found that the failures in gas power plants were due in a majority of cases to the inefficiency of the cleaning apparatus attached to the producer, and which mostly consists of a few vertical tubes filled with coke, or of a fan having a nozzle for water injection. If I may be allowed to advance for the consideration of American engineers a suggestion which will save them a great deal of experimenting, and much time and money, which we had to spend before arriving at the present state of complete success, it is this: that they avail themselves of our experience and adopt such apparatus as have given satisfaction in gas engine practice, even for very rich fuels, that is the Theissen centrifugal gas washer, which can be pronounced as efficient when viewed from all such points as contribute to the industrial economy of a heat power plant.

In conclusion, will you allow me, Mr. President, to take this opportunity to express what I think is the sentiment of the majority

of German engineers, i.e., the desire that the relations between this distinguished Society and the Society of which I have the honor to be a member may in future no longer be confined to the exchange of written communications only, but may be supplemented and intensified by the exchange of visits, lectures and papers, which members of the respective societies may be called upon to personally present before each other. We think that this will greatly serve to bring the two nations to an even better understanding and appreciation of their respective merits and industrial achievements than is existing now, and that it will have on the special science of engineering a similar good effect as had, in a general way, the exchange of professors from Harvard and Berlin Universities, and which by the donation of the Theodore Roosevelt Professorship received such a brilliant inauguration a few weeks ago.

*Mr. Edward W. Parker.**—I do not think that there is much that I can add to what Professor Fernald has already presented to the Society; in fact, I did not know until last night that Professor Fernald had intended that I should be called upon to discuss the paper at all.

One item suggested by Professor Fernald's notes this morning in regard to the absence of the use of the economizer in our work on bituminous coals and lignites should be explained. I was in Philadelphia a few days ago and was told by one of the engineers of R. D. Wood & Company that in the use of anthracite it was absolutely necessary to use the economizer for heating or for use in the producer; and as Professor Fernald has stated the producer we are using was constructed really for anthracite coal and not for bituminous coals and lignites.

I will have to ask Professor Fernald to excuse me for calling attention to an error in his figures. He states that we had produced gas having a calorific value of 288 British thermal units per cubic foot. This should be 188—which is considerably higher than what we have been able to obtain from bituminous coals of the best grade.

We started this testing plant in St. Louis under a law making an appropriation for analyzing and testing the coals and lignites of the United States in order to determine their full values and the most economical method for their utilization. We did not

* Of the United States Geological Survey.

have much difficulty in getting men for certain branches of our work. We had a number of men trained in boiler or steam raising tests; and we think we got as good a man as there is in the country when we got Professor Breckenridge of the Illinois University.

Professor Lord, who has probably done more work than any other chemist in the United States on the analyzing of coals, has had charge of our chemical work, and we have been making the most complete series of chemical analyses of coal, both proximate and ultimate, that have been made in this or any other country.

The hardest man to find was a producer man. As Professor Fernald has stated, there had been no tests on the gas-producer qualities of coals as there had been on their steaming properties. There were no laboratories and no experts. Professor Breckenridge suggested Professor Fernald as the proper man to take up that line of work. It will be readily understood that there was a good deal of preliminary and preparatory work to be done in addition to the actual investigations before he arrived at the results that he has been able to place before you to-day. Altogether I think you will agree with me that we got the right man for that particular work.

Mr. A. A. Cary.—I am sorry that I was unable to be present during the reading of Professor Fernald's paper and also to hear the remarks following its presentation, as this paper deals with a subject of constantly increasing interest to me. All progressive power and fuel users will certainly become more and more interested in gas-producer work within the next few years as its superior economy when used with the gas engine in the production of power and its great possibilities when used in connection with furnace work are matters of too great importance to fail to receive proper appreciation.

This paper presents much valuable information, which has not existed before, and it will doubtless be of considerable assistance to those who are taking up this comparatively new line of work.

I notice, however, that it deals more especially with the gas-producer used in connection with engine work; but the gas producer's other field of usefulness, that is, in connection with furnace work, is not less important, although this application is by no means as well developed.

Only those who have used natural gas and illuminating gas for heating purposes can begin to appreciate the possibilities of producer gas as a fuel, but on account of its calorific value being so

much less that that found in these other gases, it will be found necessary to use producer gas quite differently from the methods employed when the other gases are used.

In this paper, Professor Fernald refers to a gas producer in which there is a gentle current of steam passing through the fuel bed the greater part of the time, and producing a quality of gas which might be classified between an air gas and a water gas, having a calorific value of about 155 British thermal units per cubic foot.

In another construction of gas producer, we find gas generated during part of its time of operation by simply passing air through its fuel bed, and this air gas passes to a separate gas holder. Such air gas has a calorific value of only a little over 100 British thermal units per cubic foot, but I have found that it may, notwithstanding its low calorific value, be used to good advantage as a fuel.

After the fuel bed, in this process, has been brought to a proper state of incandescence by this air passage, the air admission is stopped and steam alone is blown through the fuel, making a water gas having a calorific value of slightly under 300 British thermal units per cubic foot. This rich producer gas is sent to a second holder, from whence it may be drawn either separately or mixed with the air gas from the other holder in any desired proportion.

We thus find three different kinds of qualities of gas coming from producers, commonly known as air gas, producer gas, and water gas.

The lower the calorific value of these producer gases, the greater is the difficulty found in burning them in a furnace with a high degree of efficiency, and this is principally due to the difficulty in obtaining a proper proportionate intermixture of the air required for combustion with the combustible portions of the gas itself.

The burner used for fuel gases is properly called a *mixer*, in which the oxygen in the air is supposed to be properly brought in contact with all the oxidizable portions of the gas before actual ignition begins.

Experience has taught me that when air gas or other lean gas is used, such ideal conditions are most necessary to obtain maximum temperatures, and this condition must be secured with the use of very nearly the theoretical amount of air required by the gas.

I would like to ask Professor Fernald and Mr. Parker whether

they have done any work along this line, and, if so, what forms of mixers have they used, and which have given the best results?

I do not know of any good producer gas mixer on the market at the present time. There is certainly a great need for such a device, and any information of value on this subject would be widely appreciated.

The form of the furnace is also an important matter in connection with producer gas fuel. We also need information concerning the value of preheating the gas and air suply.

Mr. S. S. Wyer.—I might mention in this connection that the American Institute of Mining Engineers took up the subject of codification of gas producer tests at their last meeting and a provisional code was presented at that time. Further work along this line will be presented at the joint meeting with the Iron and Steel Institute of London next July. The American Institute of Mining Engineers will not take up this matter in an official manner, but I am sure that some members of that institute would work with a committee that this Society might select to bring in a report on this subject.

The President.—In the matter of the preparation of the code the thought has been before the council or rather before the secretary, and I will ask the secretary to say a word on that matter.

Prof. F. R. Hutton.—It is perhaps no secret that I have been interested in getting Professor Fernald to present this paper, in the hope that it might lead to a resolution urging the formulation of methods of testing gas producers.

I was associated with Professor Fernald in some of his early professional work, when we had a good deal to do in arranging satisfactory methods for testing the gas-engine, and one of the best early methods for a complete engine test in this class was originated by the author of this paper.

In my own opinion it would be greatly to the interest of the profession if the Society, through an expert committee, could get into the procedure of testing so early in the game that the work which is bound to be done on the gas producer by those commercially interested in it should be done in such a form that it would be useful not only to those who want to buy, but those who want to make and those who want to use them.

There is no question but that great service has been done by the Society with its standard codes of procedure respecting other lines of testing. It will be greatly to our advantage if papers reporting

such tests before the Society should report them in codified form for comprehension and for the building up of sound professional standing.

I believe that we are on the threshold of very wide extension of the gas producer as a source of heat and energy and have been greatly interested in what Professor Fernald has done in the metallurgical field and what Mr. Cary has reported from the review of furnace work.

These discussions serve to widen the scope of the problem and to increase my interest in wanting the Society to take the question up.

What would suggest itself to me would be that the meeting should present a resolution to the Council asking them to consider the question of the appointment of a committee to formulate a code for the testing of gas producers, along lines parallel to what our committees have done in codifying the methods of testing boilers and engines.

If it be in order, therefore, I move that it be referred to the Council with power to consider the question of appointing a committee under the by-laws to formulate a code of procedure for conducting tests of gas producers.

This motion being duly seconded the Chairman put the question and announced that the motion had prevailed.

*Prof. R. H. Fernald.**—Mr. Cary asked whether we had investigated the question of mixers for gases of low calorific value. We have been obliged to confine our attention to testing the different bituminous coals and lignites, and have been forced to postpone incidental investigations, like the one mentioned, until the main problem has been more carefully studied. The amount of work involved in the simple testing of coals is surprising. We have been making these fuel tests for a little over a year—the tests began about the 1st of October, 1904—and we still feel that we are only in the infancy of the work.

We are hoping, if Congress should favor us with further appropriation, making the plant more or less permanent, that we may be able to take up many investigations that are closely allied with the problem that has thus far commanded our entire attention. Among the most desirable problems before us are those involved in a long series of tests of the same coal. At the present time we test one coal for sixty hours only and then hasten on to the next.

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* Author's Closure, under the Rules.

During the early operation of the plant any coal containing more than one per cent. sulphur was carefully set aside, but now we take the coals as they come. It developed at an early date that more or less sulphur was passing the purifier and entering the engine cylinders. Investigations by the chemists showed that purifiers containing oxidized-iron filings and shavings were fairly efficient for coals containing little sulphur—one per cent. or less; but it was found that for coals containing larger percentages of sulphur the purifier was exhausted after about six or eight hours. Mixtures of lime and shavings were tried but with little success. As a result of these investigations the purifier has been discarded and the gas, carrying its full percentage of sulphur, has been charged directly into the engine cylinders. This method of operating has been going on for many months and no ill effects have been discovered, although coal has been used which runs as high as 8.1 per cent. sulphur.

The operation of the plant has been a process of development. We were, as you see, a little nervous at first, but now we do not hesitate to handle any coal or lignite that is obtainable.

During the early operation of the plant all the coal was put through a $1\frac{1}{2}$ inch crusher. At the present time we feel that the operators of producers desire to use the coal as they buy it in the open market—3 inch coal, or whatever it may be—and we have consequently been testing the coal, for the past few months, as it arrives at the plant directly from the mine.

Attention has been directed to the fact that the plant is operated continuously for 120 hours each week. Although this run is of sufficient length to establish confidence in the gas producer and engine, yet we are contemplating an endurance run in the near future—simply to demonstrate the possibility of running the plant at will for any length of time desired.

One feature of the plant as installed was the economizer, used for preheating the air for the blast. A series of experiments has shown no effect upon the chemical composition of the gas or upon the efficiency of the plant when air at ordinary atmospheric temperature was substituted for preheated air. As a result the economizer, as an economizer, has been discarded, and the construction of the plant again simplified.

Other modifications and changes are under investigation at the present time, the most improtant from an economic standpoint relating to the utilization of slack coal in producers.

No. 1101.*

THE REALIZATION OF IDEALS IN INDUSTRIAL ENGINEERING.

BY H. F. J. PORTER, NEW YORK, N. Y.

(Member of the Society.)

1. In the specialization process evolved by the severe competition in the industrial world during the past half century, engineering has possibly suffered as much cleavage as any of the other professions. Our section of Mechanical Engineering was one of the earliest results of this action, and through the fostering influence of the technical school which had come into existence almost simultaneously, it prospered and rapidly assumed characteristic features which were readily distinguishable. Seeming to include the essentials required by manufacturing industry which was steadily growing and at the same time becoming continuously diversified, this section in turn responded to the pressure of the refining process, yielding many sub-divisions.

2. One of the most recent of the latter developments has been accorded considerable attention by this Society at its recent meetings; although, as its scope and limitations have not as yet had time to become defined, it does not seem to have reached an exact status. Its adherents, therefore, responding to their individual preferences or qualifications gained by personal experience have individually assumed titles illustrative of the line of their immediate practice, such as Industrial, Commercial, Organizing, Executive, Manufacturing, Production, Counseling and Modernizing Engineers. The Engineering Index classifies literature written by these men under the caption of Industrial Economy. Colleges have established departments intended and destined to embrace the fundamental principles of the new profession under the name of Schools of Commerce.

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* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

3. Some of the experts in this field devote their attention to perfecting office and accounting systems and making methods of organization and management more efficient, while others are engaged in the refinement of processes of manufacture and shop practice in order to simplify and increase production and reduce its cost; and inasmuch as office and shop are very intimately related these lines of work frequently cross and overlap each other.

4. These men are doing work as to the necessity of which there is no question. They have come into existence in response to an urgent demand, and the efficacy of their work in meeting it is evident wherever it is performed. Their efforts are directed towards the scientific study of problems in office and shop practice the economic value of which has been heretofore overlooked. There is as yet no book which treats of the whole field of their activity, although there are many monographs covering the special features of their practice. There is, however, in these latter writings an acknowledgment of the basic fact that, to secure the ultimate efficiency of the working organizations in both office and shop, the human element in each must be accorded recognition.

5. It is this feature of the broad field of what I shall here term Industrial Engineering that I desire to bring to the attention of the Society in this paper, and I do not think the subject has been referred to except parenthetically at any previous meeting. We have insensibly drifted away from the patriarchal system of management which existed as late as twenty-five or thirty years ago, in which the owner of a shop was manager, superintendent and workman, who knew little about shop organization but associated with his men on intimate and cordial terms. As manufacturing plants have grown in size and the workmen have differentiated their trades, a type of military organization has gradually developed in the shops in which the manager, superintendent and foremen are officers who instead of being expected to take part in turning out the product of the plant are, on the contrary, experts in shop management, and see that the machine of which the workmen are parts is kept in proper working order. In this way the old personal touch between master and men (which terms still adhere in English shops) has disappeared entirely. But human nature has not changed. That element still exists in the organization, and if the full strength of the latter is to be secured it must be accorded recognition. The writers above mentioned say that no matter how perfect may be an accounting system, if it is carelessly

or unintelligently applied, it becomes worse than worthless because it will be misleading in its results; that no matter how perfectly suited to their purpose the machines or efficient in quality or shape the cutting tools, unless they are skillfully manipulated the product will be unsatisfactory, and so in either case the outcome of an enterprise possessing mediocre talent in its working forces will be jeopardized. In order then to insure a successful issue to a manufacturing venture we must look beyond the mechanical assets to the qualifications for efficient service possessed by the organization of supervisors, clerks and operatives; and it will be found that its measure of success will be determined by the standard of excellence of those qualifications.

6. I do not wish, however, to be understood as underrating the value of a high grade mechanical equipment, for if success is to be attained the equipment must be the best of its kind for the purpose. But I do wish to emphasize the greater necessity of possessing as highly efficient an organization as can be secured; for a capable organization can make the best of a poor equipment and produce good results whereas an incapable organization will not only fail to make a fine equipment productive of good work but probably in a short time will destroy the equipment itself.

7. In his paper on Shop Management presented at the Saratoga meeting in June, 1903, Mr. Fred W. Taylor quotes the attitude on this point taken by Mr. Carnegie, whom he designates as having been one of the most successful manufacturers of this country, and therefore to be considered as an authority on organization and management. Mr. Carnegie, when asked by a number of financiers whether he thought that the difference between one style of organization and another amounted to much, providing the company had an up to date plant properly located, said in effect that should some great catastrophe destroy all of his mills but spare his organization, which had required many years to perfect, he might be inconvenienced temporarily but that he could depend upon his organization to re-establish his business. If, however, he should lose his organization, even if his mills which were the best in existence were left intact, he would not have time nor strength to rehabilitate himself in the business world. Just as we have, for instance, recently seen it demonstrated that opposing armies and navies may have exactly the same guns, but that the side which has behind its guns the men of superior physique, character, intelligence and skill will win the battle, so also it

has been proven that it is not the tool that determines either the quantity or quality of product, but the qualifications for efficiency possessed by the man behind the tool who controls and directs it.

8. The lesson taught generally by the foremost enterprises of the day, in their respective industries, is that their success has been largely due to the efficiency of their working forces. Thus in recent years the importance of possessing a high grade organization has impressed itself more and more upon the mind of the business manager so that at the present time this possession is considered, by those who are most competent to judge, by far the most valuable of all commercial assets.

9. Considering for a moment the functions of the organization, together with its circumscribing limitations, it must be recognized initially that if one man autocratically dominates it its scope of expansion can never be any greater than what he can himself devise. Also that the character of work which may be realized from the organization as a whole will be no better than can be produced by the individuals that compose it.

If honest workmanship is desired, honest workers will be necessary for its accomplishment; if quantity of output is expected, skill and enthusiastic devotion to duty must be possessed by the operatives; if improvement in processes and perfection of product are to be hoped for, the workers must have intelligence and be given suitable opportunities for its expression; if the organization as a whole is expected to grow, not only in size but in strength and character, facilities must be offered for the improvement of the individuals and inducements made for taking advantage of them; if untiring aggressiveness against competition is desired, absolute confidence in the enterprise and its product will have to be developed in the organization, and if harmonious co-operation between departments and management is to be effected, a spirit of mutual helpfulness must be imbued in the individuals comprising them respectively.

What then are the qualifications with which a high grade organization should be endowed in order to attain a high standard of efficiency? In as much as, after all, an organization is but a collection of human beings, it stands to reason that if as an entity it is to be of high grade the units of which it is composed must also be of that grade. As, however, no two organizations are governed in their formation by similar conditions, each case presents a problem *sui generis* for the manager to solve, but every case will be

found ultimately to consist in a study of human nature in its many phases, and this, we have been told, "is the proper study of mankind."

10. The attributes which human beings possess in common fall into three classes; *viz.:* physical, moral and mental. Those belonging to one or another of these classes may predominate in men following different pursuits in life, but in any manufacturing enterprise it is evident that the physical attributes are the most important. No matter how moral or intellectual a man may be, if he is a weakling, if he has not health, he cannot be an efficient part of an active organization. First of all, therefore, the members of the organization must have health, they must be strong and vigorous.

11. Next he must be of high character, for no matter how healthy or intelligent a man may be if he is immoral, *i.e.,* if he is dissipated, if his time outside of his working hours is given over to intemperance, gaming, or other forms of vice, his condition physically and mentally while at work cannot be such as to qualify him as an efficient member of a high grade organization. His physical and mental powers are gradually affected, his sense of responsibility weakens, he becomes irregular in attendance, careless in attention to his duties and cannot be depended upon.

12. And finally, it is evident that given a man of good physique and strong character, the higher his intelligence and skill in the direction of his duties the better qualified he will be to accomplish his daily tasks, but it is also evident from the preceding considerations that important as are these latter qualifications they must be subordinate to the other two and considered last in relative importance.

13. Now, I claim that the ordinary system of management in selecting and maintaining an organization, reverses the order in which the value of these qualifications for membership is estimated.

The questions usually asked an applicant for employment are:

First. Have you had a college education?

Second. What has been your practical experience?

Third. Have you ever been discharged? If so, what for?

Fourth. Give a list of people for whom you have worked.

Occasionally: Are you a drinking man?

But the self-estimate of what constitutes a drinking man makes the answer of little worth, and scarcely any interest is aroused by the amount of evasion displayed by the applicant.

14. Although we have seen that the human asset is of greater importance in an industrial enterprise than its mechanical possessions, I think I can safely say that infinitely more pains are taken in obtaining a machine or tool, to see that it is covered by detailed specifications and inspected during construction and before acceptance than is given to securing the man who will be put in charge of it and held accountable for results from it afterwards. Experts from at least one department and frequently from several are consulted, and much time and thought are devoted to the consideration of the machine, and after it has been secured and become a listed possession considerable care is devoted to lessening its subsequent depreciation and it is never discarded without protracted debate and calculation. On the other hand, we rarely hear of a recognized labor or employment department in a shop, and when there is one it is seldom intelligently administered. Usually little care is taken in the selection of the men, and no thought is given to their improvement during their time of service. They are taken on and laid off without a thought of consideration for their welfare or convenience, and they frequently live with a sword of Damocles hanging over them, continuously expecting on any pay-day to find their discharge ticket in their envelope. No men can work at their best efficiency under these conditions.

15. There is one marked difference between machines and the men who operate them which should be noted here. Machines, no matter how well they may be cared for, depreciate from five to ten per cent. per year owing to the advent of other and improved machines; while men, if they are properly cared for, may appreciate in value several hundred per cent. in the same time. Yet I maintain that as a general rule machines and tools are nurtured, fostered and preserved long after their period of usefulness has expired, while the permanency of service of the men of the organization who operate them receives comparatively no consideration whatever. It cannot be expected that the men of an organization will show any spontaneous enthusiasm for, interest in or loyalty to a management that openly displays so little interest in their welfare.

16. I think these statements are facts based on the aphorism that the exception proves the rule. For there are exceptions so conspicuous as to make the rank and file of manufacturing concerns in this regard seem commonplace.

17. I do not, however, wish to find fault with the care which is

generally taken in the selection and preservation of equipment, nor do I wish to be considered as casting reflections upon the lack of consideration for the men that has existed in the past.

18. Our ideas about the treatment of men by those in authority have undergone a vast change in the past few years. We are, for instance, only now beginning to realize that it pays better to educate our criminals than to keep them in solitary confinement; and our state money is being directed towards making our penal institutions places where the inmates will receive instruction and be surrounded by influences which will make them good citizens after incarceration instead of imposing punishment upon them for the useless object of satisfying revenge; which treatment, having been found to develop resentment, tends to increase rather than diminish the criminal proclivity.

19. Huxley says: "There are two opposing methods at work in the government of the world, respectively, the ethical and the cosmic. The practice of that which is ethically best involves a course of conduct which in all respects is opposed to that which leads to success in the cosmic struggle for existence. In place of thrusting aside or treading down all opposition it requires that the individual shall not merely respect but help his fellows. Its influence is therefore directed not so much to the survival of the fittest, but the fitting of as many as possible to survive."

20. The tendency of the times now is directed away from the cosmic and towards the ethical method of management in industrial and commercial affairs. We are beginning to realize that no success counts which is obtained at the expense of others; that no triumph is a real one which is attained by pulling others down or by keeping them down; in fact, that no life is worth very much which does not consider the welfare of others. The United States Government, for the first time at any industrial exposition, exhibited at St. Louis last year a large number of photographs and pamphlets descriptive of methods of industrial betterment adopted by leading manufacturing concerns in this country, thus showing the present trend of thought in this direction. The United States Bureau of Labor in its report for 1904 dwells especially on the increase in the number of concerns which have taken this advanced stand. Teachers of political economy in colleges, leading clergymen, lawyers, statesmen everywhere are advocating the adoption of ethical principles in business dealings. President Roosevelt voices this sentiment in his advocacy of giving the other fellow

"the square deal," which after all is only a modern way of expressing the Golden Rule, which political and social economists all say is the most practical rule of life ever enunciated.

21. The study of cause and effect in the example set by those concerns who have applied these principles has demonstrated that they pay in a financial way. This latter fact comes as a surprise to many, but so also did the fact that model tenement houses pay better in the long run than those of the old type. That establishments that have adopted these methods are continuing to apply them and are extending their application is significant that they are pleased with their results. On these general principles is founded the work which is the function of many industrial and civic betterment associations in this country and abroad. One of these, the Institute of Social Service, established in New York in 1898, collaborates with similar institutions in the principal countries of Europe and in Japan and keeps on file photographs, drawings, pamphlets and books, illustrating and describing schemes of industrial betterment, plans of industrial villages, improved housing for employees, methods of heating, ventilating and lighting shops, means of preserving health in unwholesome occupations, methods of preventing accidents by machinery and all cognate subjects. The failures as well as the successes are exhibited as warnings and examples respectively.

22. The National Civic Federation has established local Industrial Betterment branches about the country with the object of encouraging manufacturing concerns to adopt the most approved methods of organization and management in the hope of bringing about a better feeling between employer and employee. An effort in this latter direction must be at the foundation of every individual movement on the part of any manufacturing concern which desires to advance along modern lines. It is only reasonable to suppose that if employer and employee work together for a common end, the enterprise will pay better than if each works for his own interest simply with the full persuasion that the other is trying to get the better of him. Co-operation must supplant antagonism to secure a full meed of success. So also of two competing concerns—the one which gains the closest co-operation of its working organization, other things being equal, will have the advantage in its race for supremacy. The same law governs the commercial supremacy of nations.

23. Mr. Alfred Moseley, a rich philanthropist of England, exer-

cised over the threatened commercial supremacy of his country by our industrial advance, brought here in 1902 over twenty labor leaders of different industries to investigate the difference between American and English industrial practice.

24. His visits were naturally confined to the representative shops in the principal cities and his report stated that his observations led him to note the superiority of our methods;—first, of wage payment, largely by one or another of the merit systems, based on the principle that a certain portion of the factory space was occupied by each workman, which space represented a certain amount of invested capital, and as the fixed charges on it were the same whether the workman did much or little the more work he did the better it would be for the company, and if he was paid properly the better it would be for him;—second, of encouragement offered to develop the initiative of the workman and to get him to suggest improvements by paying a premium on them;—and third, of having the individual workman operate as many machines as he could handle, and when new and more efficient machines come on the market the scrapping of the old ones, although they may be comparatively new.

25. Mr. Carnegie has coined the happy simile that successful industry is a partnership of Labor, Capital and Brains; like a three-legged stool, the legs of which are of equal strength, no one superior to the other. And Mr. Moseley emphasized the advantage that we obtained in our works by encouraging the workmen to use their brains, and by developing and utilizing the enormous amount of expert knowledge possessed by the workmen which was not only totally neglected in England but absolutely suppressed by the attitude of the foremen, who feared that their positions would be in jeopardy if a workman showed that he had any originality.

26. But Mr. Moseley and his party did not have time to stray very far from the beaten path, and saw principally the large and representative and successful plants. There are thousands which have not as yet advanced to the state he has described.

27. Influenced, however, by the example of the leaders in thought and action, many of the latter have recently experienced a change of sentiment and sought the co-operation of their employees. The change of attitude from one of mutual suspicion to one of mutual respect and confidence is not however effected either readily or rapidly. Traditions cannot be destroyed by a simple dictum. Misunderstandings occur in the transfer which delay or offset the

proceedings. Some concerns have become discouraged and have discontinued the attempt; others, however, have succeeded and are proud of their achievement. There is a right way to go about it and there are many wrong ways. Care must be taken that the right kind of people representing the management and the men are selected to accomplish the result.

28. Alexander Hamilton said more than a century ago, "A government must be fitted to a nation much as a coat to the individual; and consequently what may be good at Philadelphia may be bad at Paris and ridiculed at St. Petersburgh." Similarly a system of shop management which is suited to one industrial enterprise may work only indifferently in another and be a complete failure in a third. This statement may seem to be trite and unnecessary, but I know from my own experience that as long as men are constituted as they are, and they have probably not changed much since the time of Hamilton, whenever unusual conditions develop in a shop under any form of management, suggestions as to remedies which have been successful elsewhere are at once made by people who are well meaning but have no knowledge of the actual situation. This is empirical, and Mr. Charles Francis Adams of Boston in a recent paper on Industrial Conference says that "in these (industrial) matters empiracy is of all things to be shunned." What is needed is a careful study by those whose knowledge of the conditions as they exist constitutes them as most competent to analyze the situation and deduce proper methods of procedure.

29. In any manufacturing enterprise a standing committee on which both employees and management are represented, which meets at stated intervals, is an excellent method of "getting together" those who are competent to act that each may see the other side of the shield and obtain the other's point of view.

30. There are frequent occasions arising when the management is anxious to know what attitude the employees would take in case a change in policy is made. On the other hand, the employees from time to time wish to lay a request before the management. In either case some temporary device is usually resorted to to meet the occasion. When a standing committee exists, however, a channel of communication is always open and each side keeps thoroughly in touch with the other on all matters of common interest. An honor system is thereby established among the employees and the discipline improves greatly. I have never failed to see a marked change come over the entire organization as it rose to its

responsibility as soon as the members felt that they were accorded recognition as rational beings and to be consulted on matters of common interest. Generally the rank and file of the working organization is considered in the same category as the privates in an army; they are not supposed to think, but to do as someone above them has planned. The usual result, as might be expected, is that they do not use their brains for the benefit of the concern.

31. All questions regarding systems of wage payments should come before the standing committee. The introduction of any of the various merit systems can be brought about in a shop with little difficulty if the subject is intelligently discussed by the committee in advance. In this country an appeal to reason is always accorded sympathetic reception, whereas the forcible application of a policy based on general assumptions meets with merited resistance. And I should say here that as all men work primarily for their support, the wage question is the one which must be settled first and satisfactorily before any schemes of industrial betterment can be effectively developed. Men work for wages, and will go where they can get the most pay. But of two concerns, each paying the same wage, the one which extends the better treatment to its employees will be favored by the applicants for employment and by judicious methods it can obtain the better class of help.

32. Some work of common interest must be developed to engage the attention of the standing committee and the sub-committees which later may be appointed to handle details. The Suggestion System now so generally adopted by the foremost concerns is a good subject for the committee to handle. This system develops in the operatives the power to observe, improves their capacity of initiative and inspires their ambition. It has besides the great advantage of being in itself a paying institution both to the employee and the company, when the latter pays well for valuable suggestions. A recent magazine article over the name of the Welfare Manager of the National Cash Register Company, which has probably gone into industrial betterment projects deeper than any concern in this country, says that this system will net his company the neat sum of $30,000 this year.

33. This system may include suggestions from the employees regarding improvements in their own conditions of comfort and work. No one knows better than the workman himself if he is not working under the most comfortable circumstances, but he will naturally hesitate to complain. Yet everyone knows that a

man when comfortable can do better work than when he is uncomfortable, and should be given an opportunity to express himself regarding his condition.

34. The system should also include suggestions regarding improvements in processes and methods of work. The operative, if encouraged to think, will soon effect great savings in the work at which he is more of an expert than anyone else who is not constantly engaged at it. I have recently seen a product considerably redesigned, the process of manufacture simplified and made much cheaper, and the company put upon its feet, largely through the introduction of the Suggestion System.

Meetings of the foremen should be held regularly, and instruction given to them in the proper handling and management of men to secure the best efficiency within the limits of fair and just treatment. It must be understood that in order to secure the best quality of work the mind of the worker must be as free as possible from worry about his position. Peace of mind cannot be maintained while conditions exist which entail arbitrary discharge from service or unexpected reduction of wages, nagging by foremen who are affected by favoritism, ineffective facilities for service, unpleasant and unhealthful surroundings, danger of accidents from unguarded machinery, or loss of life in inflammable buildings inadequately supplied with means of escape. On the contrary, credit should be given to each employee for every effort to do his part in advancing the interests of the company. A little encouragement at the right time will do much to arouse ambition and enthusiasm, without which nothing of moment is accomplished.

35. I have already shown that health, character and intelligence in the order named are the essentials to be sought for in selecting an employee, and when found should be fostered and improved. First, then, efforts should be made to raise and maintain at a high grade the standard of health of the organization. Good health means capacity for work both in efficiency when at work and through increased regularity of attendance. Irregularity in attendance is one of the banes of good management and it can be improved only by a careful investigation into its causes and application of the proper remedy. Mutual Benefit Associations are now quite general; but those where either a doctor or trained nurse is engaged to instruct the employees how to prevent sickness as well as to administer prompt and effective treatment during incapacity

for work are effective means of raising the regularity of attendance. The latter can also be effected by the payment of a bonus annually to those whose presence has met a certain percentage of regularity. Instruction how to live properly, how to cook simple food, how to eat, bathe and sleep, and warnings against the use of patented nostrums will tend to preserve the health of the organization and increase its efficiency. Employees who get a warm lunch day in and day out, year after year, will do better work after it than if they had a cold one, and facilities for warming the lunch can very readily be supplied. Means afforded for exercise and amusement in the open air at noon, especially when the occupation is confining and sedentary will also have the same effect. The character of the individual should be a no less important consideration in the mind of the employer than his health. It is very necessary to know whether or not the employee can be relied upon in the quality of his workmanship and in his effect upon others in the organization. Some managers say that they are not interested in knowing what their employees do outside of working hours. Some employees say it is none of the manager's business what they do in their own time. When however what the employees do in their time affects what they do in their employer's time, then it may be to the latter's interest to look into such matters. What the employer should desire is to have as few changes take place in his organization as possible. Now it should be remembered that in every factory there are apt to be many operatives who, because they are strangers in the locality or from reasons consequent to their condition and circumstances, have no opportunities for social recreation and pleasures and no places to go to after working hours. Therefore being more or less creatures of circumstances, they drift into associations and habits which may be debasing in their tendencies. This sort of thing leads to a lowering of the status of the organization through association, if they remain in it, or to an unsettled state if there are many changes due to constant removals. The social instinct in man causes him to seek companionship; and if opportunity is given the members of the organization to attend lectures and other social gatherings which are attractive and at the same time elevating in their tendency, the whole fabric of the organization becomes homogeneous, of high grade, and remains intact.

36. And, finally, the education of the employee should receive the attention it deserves. Early opportunity should be seized to im-

prove the mind of the employee as he grows. The wider his knowledge, the better he will perform his duties.

Apprenticeship schools in operation during working hours under the charge of a trained teacher are effective means of developing the mentality of the organization and at the same time of getting in close touch with the employee early in his career, and the longer he stays with the concern and the more money is spent on his improvement the more valuable he becomes as an asset and the greater the effort should be made to retain him in the organization. Schools should be established in the factory during evenings for the purpose of helping the employee to advance in the organization. Care should be taken, however, not to coddle the organization. Coddling engenders weaklings. Extending opportunities so that the employees can help themselves develops self-reliance, self-respect, and, at the same time, regard for the management. Such a policy promotes a strong and healthy organization. Employees are quick to feel any interest taken in their welfare and as quick to reciprocate. To increase the efficiency of an organization so that each employee is not only a passenger in the enterprise but effectively pulls his own weight is the function of this new field of Industrial Engineering. To do this with a spirit of honesty and fair dealing between employer and employee is a delicate mission. It embraces a wide field of activity in the world of advance in which industry is the leading exponent. Apart from the practical results obtainable by the foregoing methods of management, lies the fact that it is by instituting only the highest methods that employers are utilizing their privilege as men and citizens to help in the march of mankind toward progress and enlightenment.

DISCUSSION.

Mr. John Calder.—In the first twenty-eight paragraphs of his paper Mr. Porter states a many-sided problem which presents itself to industrial engineering managers, viz.: Apart from the wage system—merit or otherwise—in use, how, in view of the increasing division of labor, the magnitude of plants and the specialization which marks industrial progress, can the unit of production—the individual worker—be best enlisted, stimulated to greatest efficiency, physical and mental, and prevented from dropping out of sight in the mass and losing initiative and ambition

as a private in one of the great regiments which constitute the rank and file of modern manufacturing organizations ?

Mr. Porter concludes a general statement of the problem with a warning against empiricism in its solution which will bear repetition. "What is needed is a careful study by those whose knowledge of the conditions as they exist constitute them as most competent to analyze the situation and deduce proper methods of procedure."

The failure to observe this caution is largely at the root of the non-success which sometimes attends the slavish copying of attractive industrial betterment features from some plant where they evolved naturally under intelligent oversight, or possibly where they are merely superfluous advertising features, and applying them hastily with extravagant expectations to new conditions not ripe for the scheme, or possibly not suited in any case for the particular experiment.

Hiring and labor-reward practice are well represented already in the Society transactions, and to place the discussion on a practical basis, a few notes are submitted herewith on the actual experience of a factory management with the detailed work of initiating and forwarding some industrial betterments such as Mr. Porter advocates.

In the work of originating either technical or social factory betterments, a standing shop committee is not usually a success. In administering well-advised external betterments, however, the more the responsibility can be shifted from officials to employees, the better for the scheme.

At the outset a clear distinction should be made between the limited but very desirable and profitable technical schemes applicable with success to the internal affairs of any manufacturing establishment and the larger number of less generally applicable schemes which are conditioned by local environment. In the latter are embraced all physical and mental recreative schemes which may be most desirable in a village or small town industry monopolizing the labor of the place, and quite impracticable or unnecessary in a large city where private enterprise in these directions is abundant.

The chart appended to this discussion shows the relation which internal and external betterment schemes bear to the general organization of a particular manufacturing concern of considerable

ORGANIZATION OF REMINGTON STANDARD TYPEWRITER FACTORY AT ILION, N. Y.

WYCKOFF, SEAMANS & BENEDICT.

- Manager.
  - Executive Departments.
    - General Office Department..
      - Correspondence.
      - Messengers.
    - Purchasing Department.....
      - Purchasing of Raw Material and Supplies.
      - Receiving Clerk.
      - Raw Material and General Supply Stores.
    - Finished Parts Stock and Production Order Department......................
      - Finished Parts Stock Room.
      - Issues all Production Orders to Factories and Foundries for Parts.
      - Issues all Finished Parts to Machine Assembling Departments.
    - Shipping Department.......
      - Stock of Finished Machines.
      - Stock of Finished Portables.
      - Stock of Repair Implements.
      - Packing Department.
      - Shipping and Billing Customers' Orders for Machines and Parts.
    - Inspection Department......
      - General.
      - Departmental.
    - Labor Department..........
      - Hiring.
      - Sweeping.
      - Yard Labor.
      - Industrial Betterments.
        - Badge System.
        - Bonus System.
        - Suggestion System.
        - Baths.
        - Recreation Park.
        - Factory Band.
        - First Aid.
      - Teams.
      - Sanitation.
    - Cashier's Time and Cost Department...............
      - Cashier.
      - Time Records.
      - Pay Roll.
      - Wage Rates and Labor Records.
      - Net Cost of Product.
      - Departmental Expenses.
    - Works' Engineering Department.
      - Power Plant.
        - Light.
        - Heat.
        - Power.
        - Ventilation.
      - Maintenance and Betterments.
        - Belting.
        - Oiling.
        - Installation.
        - Carpentry.
        - Building.
        - Painting.
        - Plumbing.
        - Fire Protection.
        - Watching.
  - Manufacturing Departments.
    - Foremen & Assistant Foremen.
      - Manufacture of Parts.
        - Pattern Shop.
        - Iron Foundry.
        - Brass Foundry.
        - Blacksmith.
        - Tinsmith.
        - Japan Shop.
        - Forging.
        - Punch and Press Department.
        - Automatic and Hand-Screw Department.
        - Annealing.
        - General Burring.
        - Brass Milling and Drilling.
        - Gray Iron Grinding.
        - Gray Iron Milling and Drilling.
        - General Polishing.
        - Screw Polishing.
        - Plating and Buffing.
        - Manufacture of Cylinders and Feed Rolls.
        - Tool Designing and Making—Experimenting.
        - Type Manufacturing.
        - Manufacturing Compound Parts.
      - Manufacture of Machines.
        - Assembling Frame, Wiring, etc., Group.
        - Assembling Ribbon Movement Group.
        - Assembling Carriage Group.
        - Aligning Machines.
        - Inspecting Machines.
        - Adding Special Fixtures.
      - Repairing Machines.
        - Repairing Old Machines.
        - School of Repairing.

magnitude which is the largest industry of a small town of 5,000 inhabitants and several surrounding villages. In the Remington Standard Typewriter factory at Ilion, New York, owned by Messrs. Wyckoff, Seamans & Benedict, the chief internal industrial betterments are the Badge, Bonus, Suggestion and First Aid schemes. Like many other manufacturers with large plants where skilled labor is required, they recognize the desirability of offering some definite and perhaps pecuniary incentive to diligence and faithfulness and some tangible inducement for their employees to remain with them.

Badge System.

The following plan, which is considered to work admirably, has been in operation about three years:

All employees in good standing whose connection with the company has remained unbroken for ten years are permitted to wear while they hold their positions a handsome service badge. The badges are marked in figures to distinguish between 10, 15, 20, 25 and 30 years' service. They are also recognizable by their enamel colorings, green, red, blue, yellow and lavender, respectively, and are much prized.

Bonus System.

The badge is not merely a decoration, but every badge wearer continuing in good standing is entitled to participate in the bonus distribution, which consists of $50.00 in gold per man semi-annually ($100 each per year), contingent upon loyal, diligent, efficient and uninterrupted service. The fifth semi-annual bonus distribution took place in the office hall of the factory on June 30th, last, when $11,450 was distributed among 229 employees. Of these, 3 had served 30 years; 2, 25 years; 28, 20 years; 101, 15 years, and 95, 10 years. To attain a bonus reward is the earnest aim of all badge wearers, who are naturally desirous also of having the same distinction at each semi-annual distribution.

Suggestion System.

The badge and bonus systems are restricted in their application by length and general quality of service. In the prizes for good suggestions, provision is made to encourage and reward thoughtfulness and ingenuity among all employees, including bonus recipients; salaried officials, including foremen, alone being ex-

cepted. The latter are expected, in view of their compensation, to give all their time and ability to the company, exceptional service from them being recognized in other ways; one in particular being the mention by name of each foreman or assistant foreman and the suggestions from him adopted during any month in the "Remington Factory Bulletin," a mimeographed magazine issued on the first of each month, and which contains, in addition, factory announcements of a special character and a diary of business incidents calculated to stimulate interest in the affairs of the corporation.

Full details of the prize system for ordinary employees are announced to them in printed bulletins and framed certificates of award are given. The certificates and prizes in gold are presented before the assembled employees in July and at Christmas, when the semi-annual bonuses are being also distributed.

The good suggestions received range over a large field of labor, clerical, superintending and manual, and vary considerably in value. They embrace saving of material, utilization of wastes, substitution of machine for hand operations, improvement of manufacturing tools and fixtures, modifications in design of pieces to economize labor and time, new ideas in the line of the product, fire protection, sanitation, improvement of the system under which the factory is organized and operated, and of the forms and routine involved. The suggestions adopted from foremen alone for the last five months are 192 in number and are all cost-reducing.

All suggestions indicate, however, that the employees are working with their heads as well as their hands, and the publication of results tends to raise the intellectual interest of the whole force.

No single award is made of less value than $20.00 in gold. Sometimes a high award may take the form of a free scholarship in a correspondence school course. It is better to mass a number of relatively small suggestions from one employee before making an award than, as is sometimes done, to make the scheme appear somewhat petty by prizes as low as $1.00.

All successful good suggestion schemes require as an indispensable prerequisite the possession of absolute confidence by the fore men and employees alike in the management and higher officials. Where the latter are new, it is wiser to encourage suggestions, as occasion offers, by actual contact with individual employees.

When mutual acquaintance is set up and confidence established,

suggestion boxes, notices, awards and certificates will form the machinery of communication, but the mere provision of apparatus may prove absolutely barren of result.

First Aid System.

An employee with medical training, who was disabled by an accident from practicing, is at immediate call in a central department, where he has clerical duties, and has charge of the First Aid Room and apparatus. Minor bruises and cuts cause hardly any cessation of work now, where formerly they would be an excuse for a day or two's absence, and a valuable accident register is kept in conjunction with first aid treatment.

External Social Betterments.

In addition to a suite of twenty very handsome bath rooms, patronized very largely on Saturdays and Sundays by those who occupy the less modern type of dwelling, a typewriter band and a fine free library building (presented to the town of Ilion by one of the founders of the business—Mr. Clarence W. Seamans), maintained by taxation, the external betterment work is chiefly recreative.

Recreative.

Great interest is taken by the corporation in the physical welfare of its employees, and special encouragement is offered to out-door sports and exercise. Only recently a baseball and athletic park was fitted up, graded, etc., and formally tendered for the use of the Typewriter Baseball League, which is composed of teams from different departments of the works. Stands were built to accommodate 700, and, as the field covers 15 acres, it is amply large to supply all requirements of the employees in the way of a playground. The company believes in play, and holds that those who work together should have an opportunity to play together. In order that the recreation shall be most beneficial, it is so organized as to bring about the participation of the greatest possible number. Therefore, no effort is made to create and support one skillful team merely to meet outsiders and supply amusement, but seven departmental teams contending among themselves for the Seamans baseball challenge trophy—a beautiful silver vase—together with their substitutes

and practicing followers, insure the sharing of a considerable amount of exercise and enjoyment throughout the works.

The suggestion and the provision of the necessary ground and stands and of the challenge trophy by the corporation was all that was needed after it was realized that the officials were sympathetic and would themselves participate. Everything else in connection with the recreation movements was successfully organized, financed and carried out by various committees of the employees, including an annual Field Day, with an attendance of 3,000 persons. A flourishing Cricket Club of 65 members offers less intense sport for the officials, foremen and older employees. The principle underlying success in such work will bear repeating. Do not copy other schemes. Study your own conditions carefully. Let the management plan, suggest and encourage. Schemes will not usually originate in the shops, but when they are started, do not officially finance or conduct any arrangements which the employees can possibly carry out themselves.

Mr. Hugo Diemer.—From the shop owner's standpoint, the ideals to be realized in industrial engineering are:

1. The producing of a marketable product which will command the highest price of any similar product in its class.

2. The producing of the largest possible quantity of this article at the lowest possible cost.

With the development of a better educated and more enlightened purchasing class it is coming to pass that the shop owner is beginning to feel that in order to realize the first named ideal, the quality of his product must be continually improving. The most marketable steam engine or machine tool of to-day is a product of much higher quality than that which was the most marketable but a few years ago. This realization has resulted in better machinery and in the employment of better designing talent, and is the reason for the creation of the positions held by a great many of our members.

The second business ideal, namely, the producing of the largest possible output at the lowest possible cost, involves not only mechanical design and equipment, but the element of human activity. The need for pronounced emphasis on this element appears to have been felt thus far by but few manufacturers. The standpoint of advantage to the shop owner is the only point of view from which the engineer can consider welfare and betterment propositions. There are several phases of the human side

of the shop that are distinctly engineering problems. Among these I would place first:

Employees' Efficiency Records.

Human activity is the most variable of all factors that enter into a study of manufacturing economy. It is certainly within the engineer's province to prepare and keep efficiency records of men as well as of machines. The machinery record of a modern shop will list every machine tool, its original cost, its expenses of maintenance, repairs, and depreciation, and such data as may be desirable regarding speeds, methods of operation, etc.

A well-conducted shop will keep efficiency records of employees in the ranks that will enable the manager to encourage and keep the best workers and weed out the bad ones. Many shops have an employment record giving sundry details as to a man's street address, previous employment, the size of his family, etc., but I have seen few of even such meager records which are kept intelligently and in good order, and which are ever studied or even frequently consulted. I have introduced individual workmen's records showing each man's successes and failures in merit wage systems, but I have found it difficult to secure the careful study of these efficiency records which should be accorded them by persons in authority. What use can be made of such records? If we discharge the inefficient workers, what guarantee have we that new men will be more efficient? This is true, but our efficiency records reveal which men need to be given a chance to do better work, and show also which men in the shop are most capable of showing them. A working man will take more kindly to a demonstrator selected from among his peers and fellows than to a speed boss or speed expert hired from abroad. It is easy to pick out in a machine shop one's best milling machine hand, the best lathe hand, the best drill press hand, and by paying a higher hourly rate while on demonstrating work, to have them serve as instructors to the less capable men. The great majority of men who have fallen below a standard of good performance will make a success of their tasks when helped in this way.

The second phase of the human side of the shop which the engineer should handle is the apprenticeship system. There have been excellent discussions before this Society on this subject, and yet the apprenticeship system in most shops is a farce.

Mr. Porter shows that the realization of certain ideals of politi-

cal economy go hand in hand with the development of a disposition on the part of the employees to become better help. To make better men and women of the rank and file is a great fundamental step. Give them better air and better surroundings and they will have better health and better dispositions.

But we must do more than this. We must have better craftsmen with better technique in their trades. It is right here that we are weakest, and it is here that we will first feel foreign competition. The all-around machinist is almost extinct—the man who knows how to get best results out of a lathe, a milling machine, a planer, a shaper, a drill press and a boring mill. Machine-shop foremen who have advanced from the ranks are as a rule men who are proficient on but one tool, and are not competent judges of the best way to do work on all the different machines even in their own departments. The great majority of the workmen in shops to-day have not had any schooling beyond the seventh or eighth grade. Their shop apprenticeship has not made them all-around craftsmen. The philanthropy expended in the erection of manual training high schools has failed to reach the great mass of working people, who never get that far along in school life. It is this great majority of our people who never get beyond the seventh or eighth grade of school that are most in need of further trade education to make them better craftsmen. There is certainly tremendous need for the establishment of real trades schools with the best obtainable really practical instructors for this great class of working people.

When we talk of betterment of employees, the above two fields of effort which I have mentioned are not usually thought of. We associate the terms "Welfare" and "Betterment" immediately with company contributions of a philanthropic nature to a great annual picnic, Saturday afternoon baseball, benefit society funds, and similar matters.

In regard to the workingman's home life and his habits, he does not want these interfered with. However, time and again workingmen have expressed themselves to me in regard to their wishes for better home surroundings. They complain to me in the same way in various cities, that civic improvements are made in those parts of a city occupied by the homes of the well-to-do. There, they say, we find paved streets, good sidewalks, shade trees, water, sewer connections, electric lights, gas and street sprinkling, while the workingmen's homes are on dark, muddy

streets; no trees, no sewers, no paving, poor sidewalks. Workingmen complain to me that manufacturers are willing to spend time and money to influence councils to secure switches and to secure election to public office of men pledged to grant their corporation special privileges, and the workingman would like to see these same influences at work to secure public betterment of his home surroundings. This phase of the question is one that can hardly be controlled by the engineer within the walls of the shop, but it is well worth the thought of the shop owner.

The greatest successes in social betterment work in connection with factories have been accomplished in shops where the work is light, such as the manufacture of breakfast foods, chocolate, cash registers, bed springs, etc. The employees in these shops are more easily reached than are the workers in the foundry, the forge shop, and the heavy machine shop. I have attended entertainments of a high order, given without charge by the owners of one of the largest shops in Chicago, and under the auspices of skilled social settlement workers. Yet they failed to reach the men. I have in mind particularly one evening when the entertainment was a lecture on the Klondike, preceded and followed by music by the shop band. With many thousands to draw from, there was but a half dozen men outside the band present. The lecture was intensely interesting to me; yet one after another the men went to sleep. There were two reasons for this: First, the men were physically exhausted; secondly, it was not the type of entertainment that appealed. When the orchestra struck up it was with vigor and abandon, and the men came to "attention" at once. This same large shop, assisted by private donations from large stockholders, instituted much betterment work, including a manual training school for the children of the employees. A minor feature that was a good one was the distribution from a truck of bottles of pure sterilized milk sold at cost. But the greatest difficulty encountered here, as in the heavy steel industries about Pittsburgh, is the handling of the adult workman engaged in heavy labor. The Pittsburgh district abounds with magnificent club buildings provided with libraries, gymnasiums, social features, etc., open to the laborer who wishes to avail himself of their opportunities, and yet these institutions fail to reach those who most need uplifting. These clubs are taken advantage of by the young college-educated apprentices, shop clerks and some few of the young men of the higher grade

mechanic class. I should like to hear of successful plans for reaching the heavy labor class, plans which have resulted in continuous and permanent good.

I wish to add just a word in regard to the efficacy of "Suggestion" systems. A suggestion system can be made of benefit in any shop, even a small one, but only if handled systematically and intelligently. The men need to have the benefits of the suggestion scheme kept continually before them. This can be done by handing out each month individual slips to every employee announcing the prizes for the coming month. Another means is the printing on the time slips that show the workmen's gains or losses under a merit wage system, of a few words calling attention to the suggestion scheme, as, for instance, "We are always glad to have our men make suggestions for changes in fixtures, appliances and tools to facilitate the work. If an idea occurs to you, write it out and put it in Suggestion Box. Prizes are offered for the best suggestions." Suggestions should be acknowledged by a personal letter to everyone making a suggestion. Awards must be intelligently made by competent judges. Suggestions adopted should be put into effect promptly. Foremen should be made participants in the suggestion scheme on a different footing from the men.

Mr. Porter has certainly presented to us a phase of the labor problem which is well worthy of careful attention by the engineer to whom is entrusted the problem of successful shop management.

Mr. John T. Hawkins.—Upon perusal of Mr. Porter's paper I recognize in it still another of the several papers read before this Society in more or less recent sessions, having for their praiseworthy object economic improvement in the industrial world; but, like its predecessors, not only does it not show us the way to the ideals desired, but if realization of these ideals is attempted, we or they are at once "up against"—to use a very expressive colloquialism—one thing more formidable than all the unfulfilled desiderata they labor so hard to present to our anxious minds.

These and similar attempts to improve industrial economics are only gaining at the spigot with the bung out, while in the face of trade unionism and its acts, professions, aims and powers.

A large proportion of those most vitally interested in industrial advancement have too long been mealy-mouthed about trade unionism; while even after some little amelioration of its worst

and most indefensible features in later years, it still stands an inexorable ogre in the pathway of industrial progress.

There is no worse part of the trade union traditions, principles and everyday practice—Mr. Gompers to the contrary, notwithstanding—than their palpable, and to a great extent acknowledged, aim to reduce "productivity" and to keep it at a minimum, not only quantitatively, but qualitatively as well. The New York Sun has recently, editorially and cogently, said: "One of the falsest arguments of unionism is that an increased individual efficiency, or productivity, would restrict the demand for labor. Its claim is that the competent must be restrained or an army of incompetents will starve for lack of work." And again, in criticism of a paper by Professor Laughlin, in the November number of Scribner's Magazine, it well says: "Professor Laughlin, in what he calls the principle of productivity, the only wise and right solution of the labor union question, refers to this as the new policy and as the new proposals." While there would be a measure of novelty in its adoption as the dominant tenet of labor unionism, there is no novelty in the principles involved in the proposition. That was stated thousands of years ago when a wise man said: "Seest thou a man diligent in his business? He shall stand before Kings." And herein lies the gist and gravamen of the generalization that, until the industrial world can succeed in transforming trade unions into instruments for the increase of productivity, instead of as now and in the past, instruments for the minimization of product, or, what amounts to the same thing, increasing its cost, all our fine spun theories and argumentative, on even practical refinements looking to the clipping off of a fraction of cost here, or improving some apparatus in capacity to produce a fraction more there, or for the introduction of some well considered shop organization with the same end in view,—all go for naught, and must continue so to do so long as the giant of labor unionism stands in the way, ready and powerful to block the road. Professor Laughlin says, "By productivity is meant the practical ability to add to the product turned out in any industry," and the Sun goes on to say: "This covers, or should cover, quality as well as quantity. The article is reducible to a proposition that unionism, to achieve a real and not a fictitious success, must make labor efficiency its standard and its goal, rather than labor monopoly, either actual or artificial. Unionism prates much about improving the conditions of the wage earner. Tak-

ing its conduct rather than its words as an indication of its real belief, unionism sees in more money and less work the only pathway to improvement. It seeks to open that pathway with the club of monopoly.

"The unionism of to-day needs nothing else so much as it needs leaders who will preach to their followers the gospel of efficiency, of 'Productivity.' The man whose daily output exceeds in quantity or in quality the product of the average of his fellow workers has that for which employers stand always ready to pay a wage which is even above the 'wage scale' demanded by unionism. The protest of employers is against the demand that the loafer and the incompetent, merely because they hold a union card, shall be paid on the basis of highest efficiency. There is also protest against artificial restriction of labor supply in given lines. Unionism fails to see how its present policy in many ways reacts upon labor itself through an increase in the cost of articles of daily need and consumption greater than that permitted by the laws of competition to employers in the form of profits."

One very important feature in the increased cost of production through the operations of labor unions, without pecuniary benefits to wage earners, and which has been given too little notice in discussions on this subject, is the enormous amount of money required to meet the expenses of unionism; dues and assessments on members to provide the union treasuries wherewith to hire halls, pay walking delegates—in these later days euphemistically called business agents—salaries of leaders and sub-leaders, and last, but not least, to support in idleness men on strikes; all these constitute a charge upon product in shape of increased wages in which the workman has no share. It is so much added to the cost of product and an exact equivalent in reduction of productivity. The price then becomes so much greater to the consumer, and as every producer is also a consumer, the wage earner is inevitably a pecuniary loser, while he assists in the nullification of the efforts of genius and capacity to reduce the cost to him of what he consumes.

Referring to paragraph 7 of Mr. Porter's paper, Mr. Carnegie is reported, as between two evils, preferring the loss of his plant to that of his organization. But Mr. Carnegie never permitted himself to be controlled by the labor unions, otherwise he would probably have preferred the loss of his organization and to keep his plant to be run by a better one.

From paragraph 8 I quote: "The lesson taught generally by the foremost enterprises of the day, in their respective industries, is that their success has been largely due to the efficiency of their working forces," and the author might have added that the efficiency of these forces is directly proportionate to the degree of freedom they enjoy from the control of labor unions.

In paragraph 9 Mr. Porter says: "If honest workmanship is desired, honest workers will be necessary for its accomplishment; if quantity of output is expected, skill and enthusiastic devotion to duty must be possessed by the operatives." Truth! self-evident truth! But how is this to be expected from men who in secret meeting not only encourage the minimization of productivity, but appoint men in your own shop whose duty to the union is to see that your men do not exceed a limit of production set, not by you nor by the individual workman, but by the edict of the union? What sort of enthusiastic devotion to duty is to be looked for in what has come to be known as the "closed shops" (a most felicitous title), for such are closed to all progress and advancement for employer and employee alike? We may imagine the devotion to duty in such a shop when the employer is obliged to tell a customer that he could not quote him figures on a certain product in his line until he ascertained from the union how many of the pieces required it would allow a man to make for a day's work, which actually happened to the writer. Mr. Porter should have added to this paragraph: "But please, gentlemen, do not expect to achieve what I have just described unless you are in absolute freedom from the control or dictation of the labor unions;" and he should add to his four questions, paragraph 13: "Are you a member of a labor union? and if so, shall you deem it your duty to your employer to do the best that in you lies, both as to quantity and quality of product, or shall you be guided in these matters by what your union may prescribe?"

Paragraph 29 describes what would be "an excellent method of getting together" of employers and workmen—in the absence of labor unions, which latter institutions the paper never once mentions; but what might we expect from standing committees of workmen in a closed shop on one side and the management on the other, if the first be governed, as they would be, in secret meetings of the unions where the management can never have a word to say?

The first prerequisite to anything like mutual conferences be-

tween workmen and their employers, having for their object their mutual good, must be the disarming of the labor unions of the club or clubs with which, as the New York Sun says, "they seek to open a pathway to monopoly." The closed shop, the cowardly boycott and its no less cowardly congener, the unfair list, so called, the limitation of apprentices, opposition to trade schools, the sympathetic strike, the persistent and traditional efforts of the labor unions to emasculate labor-saving devices and methods, are some of the clubs which must be wrested or reasoned out of the hands of wage earners before we can hope to improve the economics of the industrial world by any such means as are advocated in this and cognate papers.

That greatest of their clubs, the closed shop, has been wielded by the unions with such effect, that the employers of the country, seeing that their very existence was at stake, have taken a stand into which they have been forced by that labor union club, and already the day is breaking, promising an early release from this worst of all tyrannies.

The industrial world must take the stand that every "shop" of industry must be an open one; then we may be able by conferences or otherwise to convince wage earners of the error of unionism as now carried on, and finally bring about that fraternity between employer and employed so much to be desired, and which would surely result in great gain to both, pecuniarily and otherwise. Then, and not till then, may we achieve such industrial progress as is looked for in the paper of Mr. Porter, and others with similar aims, which have been read before this Society, it will be worse than useless to rack our brains over any form of industrial advancement which does not first squarely face the very discordant music of the trade union band. Until this kind of a stand is taken we may not hope even to approach a realization of our ideals in industrial engineering.

Mr. W. S. Rogers.—I must plead to an emphatic coolness of interest respecting the so-called ideals in industrial engineering. They make the most valuable sort of reading advertisements ever given to a magazine artist, but I have little use for the whole affair.

I am interested in a factory, beautifully located, 1,200 feet above sea level, three miles from one of the most attractive summer resorts in Connecticut, and an ideal place for a tired working

man to enjoy life. I want a good machinist and advertise for him. Of a batch of 125 replies, perhaps sixty men come to see the place. One-half of these go back within two hours, ten stay a few days, and the same experience repeats itself day after day. When I try to find out the matter, I am told that country life is too lonesome, or that there is no place to go in the evening, or they don't like local option, or there is no place to exhibit attractive clothes. Another set don't like the houses, the snow is too deep, the well is out of doors, and the coal does not come up in the elevator.

This business of idealizing the surroundings of the industrial engineer keeps a lot of fellows busy, but if the modernizers and idealists had begun twenty-five years ago putting this surplus expenditure on wages, the working man would to-day be able to buy his own home, would have educated his children, and would have been a free and independent citizen, which the framers of our Constitution intended. Instead of that he is forced into becoming a subject of paternalism, and we who want to call ourselves "Captains of Industry" are trying to tell him how he must live and where and how he must sleep, and a lot of other stuff that he knows as well as we is "none of our business."

Mr. O. K. Harlan.—In Mr. Hugo Diemer's discussion he called attention to the fact that social betterment was most applicable to shops doing such classes of work as manufacturing cash registers, typewriters and such light products, but that it did not seem to succeed so well in shops doing heavy work.

In reply I would like to cite to him the good work which has been done and is continuing to be done (for it is now long past the experimental stage) by means of Young Men's Christian Association organizations among railroad men. These men are certainly engaged in a class of heavy and vicarious work quite different from that in cash register and typewriter plants.

I have observed the work of the Young Men's Christian Associations in this field for a number of years, and to one interested in the uplifting of industrial workmen, it will be worth their while to investigate this plan, if they are unacquainted with it.

Much could be said with regard to the class and variety of men and occupations with which it has to deal—not only with men directly on the road, but with shop men engaged in heavy, dirty work, and with office men—but I will not go into detail now. If, however, further information is wanted, I shall be glad to point out many good live organizations which are doing successful work

at a very economical cost, but still the fees assessed the members and the conditions for admission are so liberal that the men pay for their privileges and are not given them gratuitously so as to make them feel themselves to be subjects of charity.

*Mr. H. F. J. Porter.**—Just one word—I will say that possibly if Mr. Rogers had a little industrial betterment up there in his country shop he would be able to hold his employees better. He acknowledges that it is good advertising and states how futile his own methods are.

In regard to trade unionism I do not know of any better club than education and industrial betterment to counteract the bad influence of unionism. The National Cash Register Company, which has done more industrial betterment than any concern in this country, was a union shop for eighteen years, and about three months ago when the Typographical Union demanded an eight-hour day there, as it did throughout the country, the company said, as they always had, "As soon as the union attempts to run this shop we will throw it out;" and they threw it out, and that shop is now an open shop, and became so without a strike, the employees deciding that they got more benefits from the company than from the union. That seems to be a very strong argument in favor of industrial betterment.

A Member.—Is it not a fact that they have had some very dangerous strikes where the men had to be paid for work they did not do, and where, after the most vicious and prolonged strike, independence was gained?

Mr. Porter.—They had one strike about three years ago which lasted about six weeks. It was as much of a lockout as it was a strike, and at the end of the lockout the employees appealed to the president to be allowed to come back, and they came back with the same conditions existing as when they went out.

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* Author's closure, under the Rules.

No. 1102.*

REINFORCED CONCRETE APPLIED TO MODERN SHOP CONSTRUCTION.

BY E. S. HUNTING, PITTSBURG, PA.
(Junior Member of the Society.)

1. The subject of shop construction is one of the most important problems that the mechanical engineer has to solve. Economy and limitation of capital render this problem one of great difficulty. Modern business methods require that money invested shall return substantial percentage of profit—consequently, it has been necessary for the mechanical engineer to devise some substitute for fire-proofed steel construction that will answer the same purpose for less money.

2. It is the writer's object to set forth a few of the advantages of reinforced concrete and to show how well it lends itself to shop construction; also to give some data on actual work of this class.

Adaptability.

3. Concrete is a mixture of sand, stone, cement and water. The increased demand for cement has caused plants for its manufacture to spring up in almost every locality. Sand, stone and water can be obtained everywhere, locally. The mixture of the aggregates can be made by very efficient mechanical devices or by the use of the most ignorant class of labor—with the same good result. This mixture when completed can be moulded to any shape or form from the rough foundation to the most artistic design of cornice or capital. The steel reinforcements are of standard sizes and shapes and are readily obtainable in any market on short notice. The tonnage of this steel work is small and of very light section and requires no apparatus to set in position.

* Presented at the New York meeting (December, 1905) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*

Strength.

4. As a structural building member reinforced concrete shows a very economical distribution of material. Steel is provided to take care of all direct tensional stresses and those shearing stresses for which the concrete is not sufficient. Compressive stresses are taken care of by the concrete. In other words, at least one-half the stress in a beam is provided for by the concrete whose unit cost is comparatively low. The discussion of the various methods of calculation used in beam design has been very thoroughly taken up by our technical journals and would require a paper in itself to thoroughly cover the ground. Most theories advanced are, however, based upon the common theory of flexure, no allowance being made for tension in the concrete. The theoretical discussion has been developed to a great extent by European engineers.

Fireproof Quality.

5. Concrete is a poor conductor of heat. Subjected to a high range of temperature, concrete gives up part of its water of combination and becomes a much poorer conductor of heat than before. Hence, there is but little danger of the tension steel in a beam giving away when well protected by concrete. After an examination of fireproof buildings in the path of the Baltimore fire by a committee of experts composed of H. de B. Parsons, M. Am. Soc. C. E., S. C. Weiskopf, M. Am. Soc. C. E., and Carl Grieshaber, the conclusion was that reinforced concrete surpassed all other materials for fireproof qualities.

Durability.

6. Forces of nature, no matter how severe, have but little effect on concrete. Subjected to severe tests of acid fumes and high temperatures, as was the case at the fire in the Pacific Coast Borax Company's plant at Bayonne, N. J., concrete showed but slight signs of deterioration. Edwin Thacher, M. Am. Soc. C. E., in a paper before the International Engineering Congress points out a number of tests that show that steel properly protected by concrete will not deteriorate.

Economy.

7. In shop construction, reinforced concrete is from 10 to 20 per cent. cheaper than a similar design of fireproofed structural steel. The largest cost item for concrete construction is the forms. It can be readily seen that any decorative work adds materially to this item—while the concrete itself costs no more in an artistic cornice than in any floor or beam—the mould in which it is placed requires the employment of highly paid skilled labor in its manufacture. Compared with slow burning types of construction, the costs vary in different parts of the country. In some sections reinforced concrete can be built for almost the same figure as slow burning construction.

Effect of Fire on Structural Steel.

8. It seems almost incomprehensible that in the modern structural steel machine shop there is sufficient inflammable material to cause a destructive fire. It is true, however, that but very little fire is necessary to make a structural steel member give way when under stress.

Fig. 1 shows a modern machine shop in the Pittsburg district, constructed of unfireproofed structural steel and equipped with modern fire apparatus.

Fig. 2 shows the same shop two days later, a mass of ruins that can be dismantled only by the use of the cold chisel and sledge.

Example of Reinforced Concrete Machine Construction.

9. Taylor-Wilson Mfg. Co.'s plant at McKees Rock, Pa. The ruling factors in the design of the Taylor-Wilson Mfg. shop were:—

First. It should be an absolutely fireproof building.

Second. It should be built at a minimum cost.

Third. Provision should be made for heavy craneways.

Fourth. The design should have some artistic value.

Fifth. Abundance of natural light.

Sketch of Foundations.

10. The shop is 160 feet long and 102 feet wide, and is carried on a series of foundation piers running down an average of 12 feet to hardpan. These foundations were put in and a fill

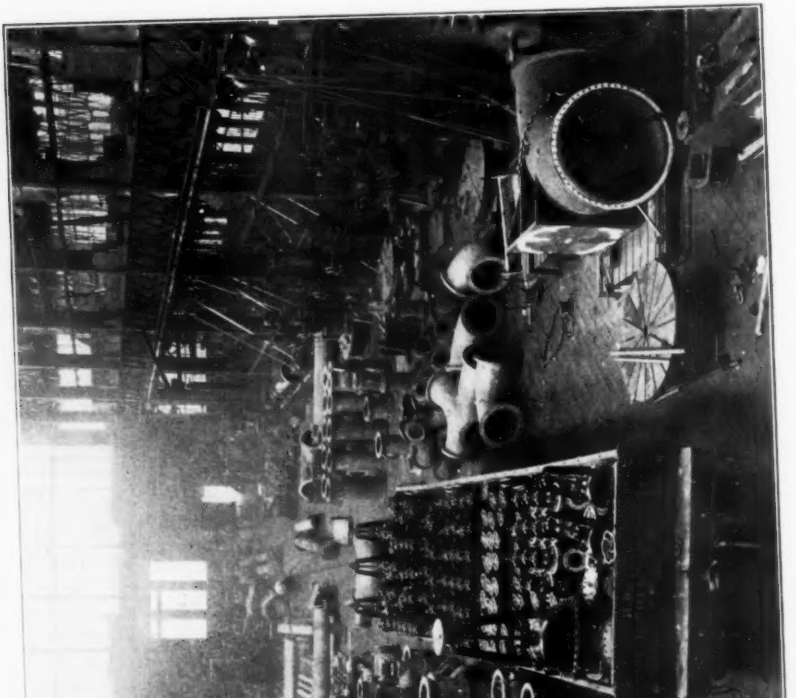


Fig. 1.

FIG. 2.

FIG. 3.

made around them. In the rear of the building, which is on swampy ground, this fill was about 22 feet. In plan the shop consists of a center aisle 51 feet 7 inches wide—2 leantos 18 feet on one side and 30 feet wide on the other.

Columns.

11. The entire building load is carried on four rows of columns. Two outside rows of 12 inches square and two inside rows of circular columns 20 inches in diameter.

12. Fig. 4 shows the method used in the construction of the moulds for the 20-inch circular columns. These moulds were formed of 16 gauge galvanized iron and were very satisfactory, giving a perfectly true and smooth surface. The column reinforcement consists of four vertical rods, to which were attached a series of hoops $1\frac{1}{2}$ inches wide and $\frac{1}{8}$ inch thick, spaced 4 inches apart. In this connection it might be well to state that the vertical rods are considered useless as far as carrying the load is concerned, and come into play only when the column acts as a beam due to eccentric loading. The theory of this design is that compression is not a stress in itself, in reality failures by compression are failures by secondary tensional stresses within the material. It has been found by a long series of tests that a column failure is always due to a tendency of the concrete to bulge, consequently the strength of this design depends, above a certain loading, upon the strength of the bands encircling the concrete. Ultimate stresses as high as 10,000 pounds per square inch have been developed. These bands are rigidly attached to the verticals for spacing and have a projecting fin that holds them to the proper distance from the form.

13. Where the column runs into the beam it will be noticed that the area is increased. The reason for this is that higher unit stresses are allowed in the hooped column than in the beam, so it is necessary to increase the area at the junction.

Beam Construction.

14. Fig. 5 shows the main beam of the building—the crane girder. This girder was designed for a 30-ton crane, and spans 20 feet between the circular columns. In section the beam is 18 x 36 inches, and has an upper flange. This upper flange takes care of the thrust due to the cross travel of the crane.

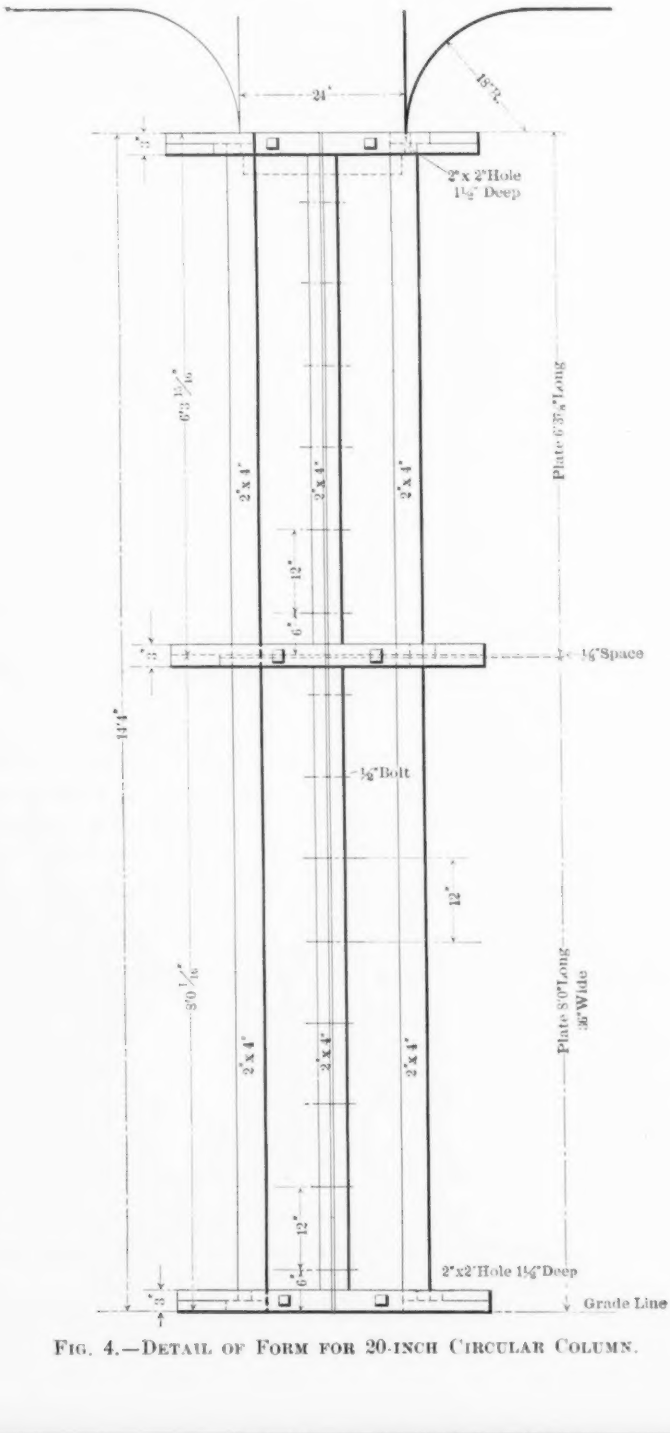


FIG. 4.—DETAIL OF FORM FOR 20-INCH CIRCULAR COLUMN.

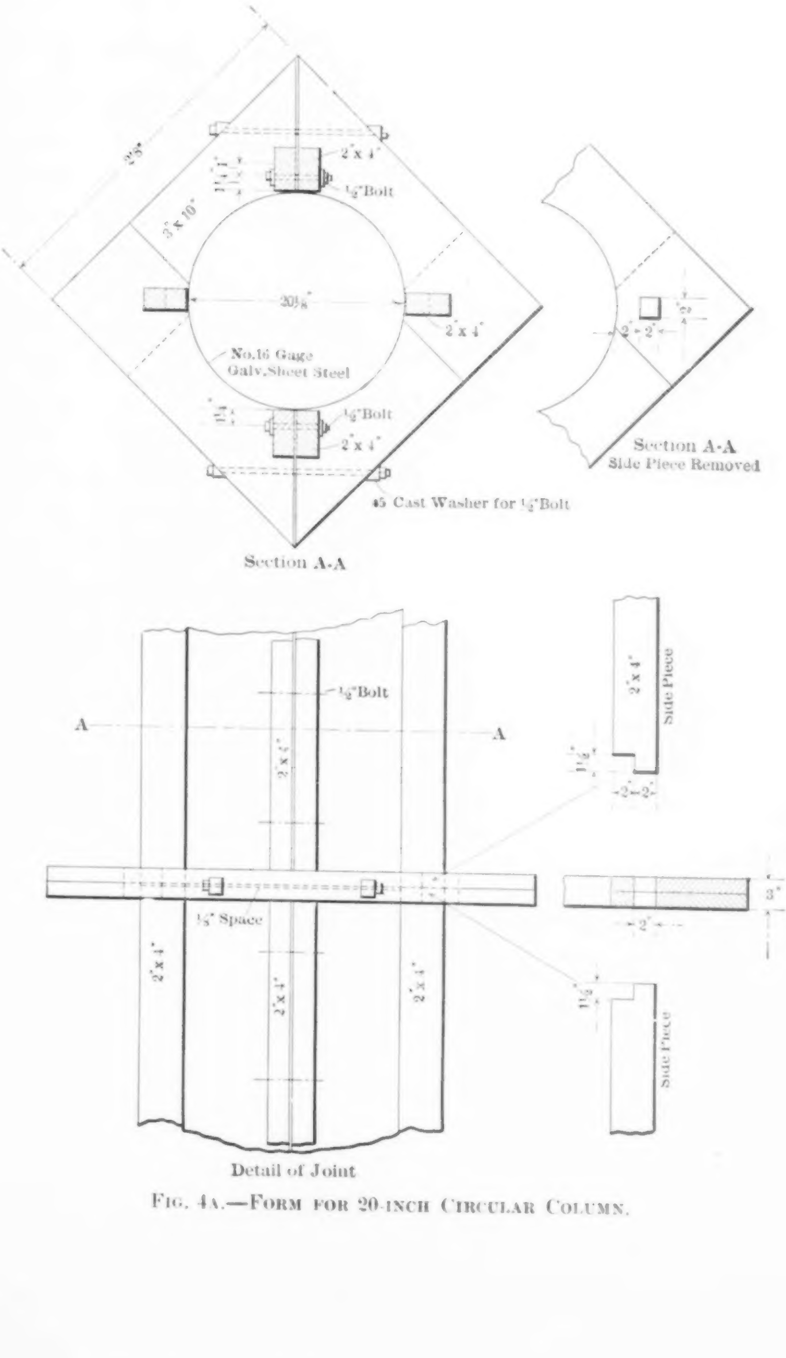


FIG. 4A.—FORM FOR 20-INCH CIRCULAR COLUMN.

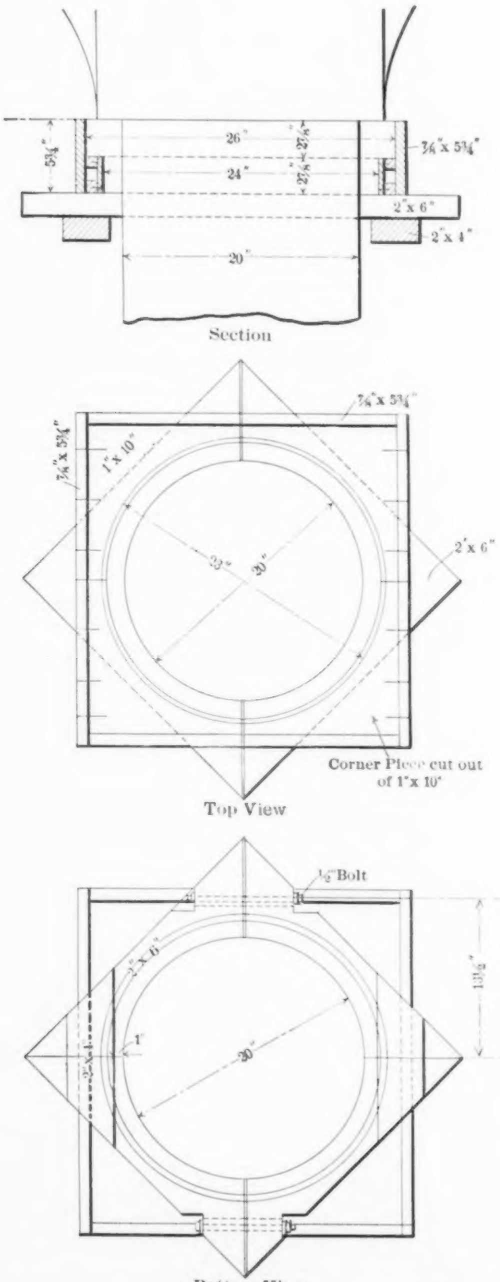
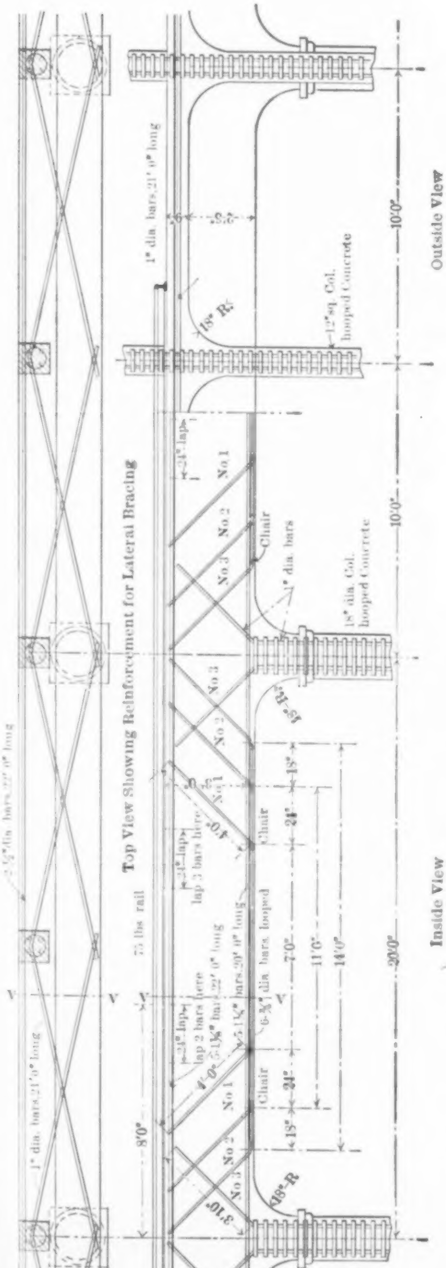
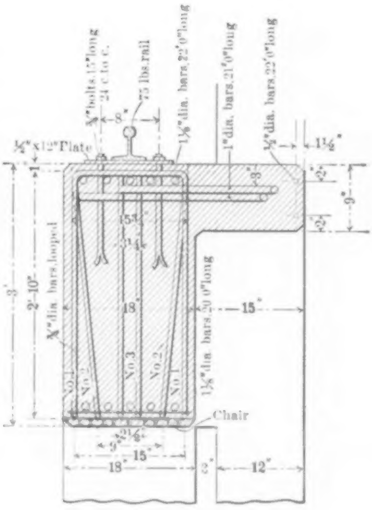


FIG. 4B.—DETAIL OF FORM FOR COLUMN CAPITALS.





Detail of Section A-A

FIG. 5.—DETAILS OF CRANE RUNWAY FOR 30-TONS CAPACITY CRANE.

15. Fig. 5 shows the reinforcement, practically in the form of a Pratt truss running continuous over the columns The bars are in the form of loops bent up at the ends to take care of the shearing stresses. The advantages of this loop are that it is self-supporting and utilizes the material economically. While a great deal of metal is necessary at the center of the span to take care of the tension, due to bending moments, these bending stresses decrease toward the supports and the same metal section is not required at the bottom of the beam, so is utilized to take care of the shearing stresses. For supporting the rods and keeping them spaced properly and away from the form a little device stamped from sheet metal is used. This spacer is notched out and parts bent down to support it—leaving openings to receive the bars. It was found to be far better than any system of concrete block support for the reason that it held the bars rigidly in their proper position.

Arch Design.

16. Covering the main aisle—spanning 54 feet—a concrete arch was constructed 4 inches thick at the crown and 10 inches at the haunches, with stiffening ribs 5 inches wide and 10 inches deep, spaced 10 feet 0 inches, center to center. The stiffening ribs were reinforced with two $\frac{3}{4}$-inch rods.

Arrangement of Steel in Arch.

17. Fig. 6 shows the arrangement of steel in the arch. This reinforcement consists of $\frac{5}{8}$-inch bars 9 inches on centers, running across the arch. Running up and down the roof, laced between the $\frac{5}{8}$-inch bars are a number of strips of band iron 1 inch by 1-16 inch, arranged so that in case the rods at the intrados and extrados act in compression there will be no danger of buckling (see Fig. 7). This design was based upon the elastic theory, using Cains' method.

Fig. 8A-8B, showing work during construction. Fig. 8B shows the status of the work, January 2, 1905, with arch roof partly completed. Fig. 9 shows an inside view during construction, crane, girder and supporting columns.

Design at the Haunches.

18. To take care of the thrust of the arch—tie rods made up of two 3-inch by $2\frac{1}{2}$-inch by $\frac{7}{16}$ angle irons were used, spaced 10 feet

Detail Showing Arrangement of Steel in Roof Arch

FIG. 6.—TAYLOR & WILSON MACHINE SHOP, MCKEES ROCK, PA. REINFORCED CONCRETE.

apart. These tie rods were bolted to two 10-inch 15-pound channels back to back, shown in Fig. 10. The channels distributed the load, due to the thrust between the rods. A light skew back casting was placed in the upper channel to act as a spacing member for the roof rods and to transfer the load to the tie rods. To take care of the uncommonly large temperature stresses that would naturally be developed in such a large thin area, expansion joints were made every 10 feet in the arch. The entire arch was constructed during the coldest winter months. Although winter construction in concrete is not commonly considered good practice among engineers, cool weather is the most advantageous time to handle this class of work. The reason for this statement is that when temperatures are low the cement and aggregates of concrete are of the smallest volume and contraction due to temperature stresses is seldom found on work carried out in the winter. Cracks seldom develop from expansion—almost entirely from contraction.

Protection from Frost.

19. It was necessary to adopt some heating system to prevent the work from freezing during winter construction. A system of heating by live steam was installed. Any system for heating concrete work that does not provide moisture as well as heat should not be considered. It is absolutely essential that concrete should not be forced to take its set quickly. Concrete that is reinforced

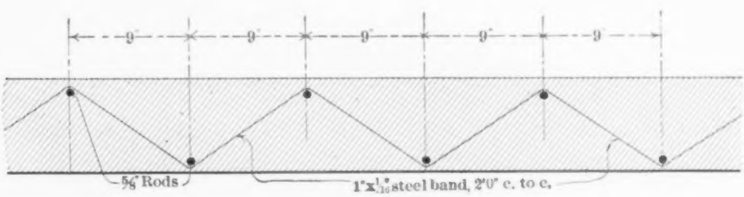


FIG. 7.

with steel should not be protected from freezing by the use of salt. The reason for this is obvious—salt has a great affinity for moisture, and moisture and air corrode steel quickly, and concrete placed in the winter is liable to be somewhat porous, and air will penetrate and the steel reinforcement will become oxidized. The heating system by means of live steam jets was very satisfactory and no bad effects were experienced from frost.

FIG. 8A.

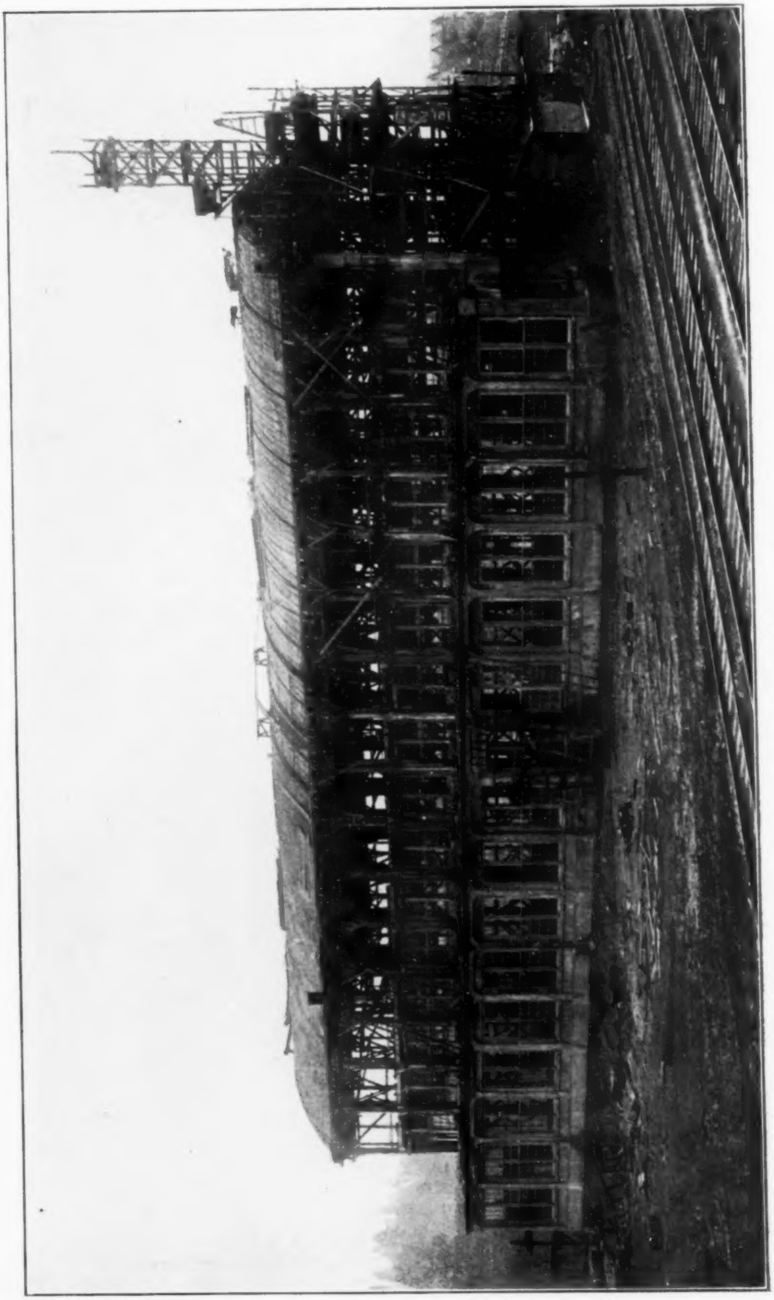


Fig. 8b.

FIG. 9.

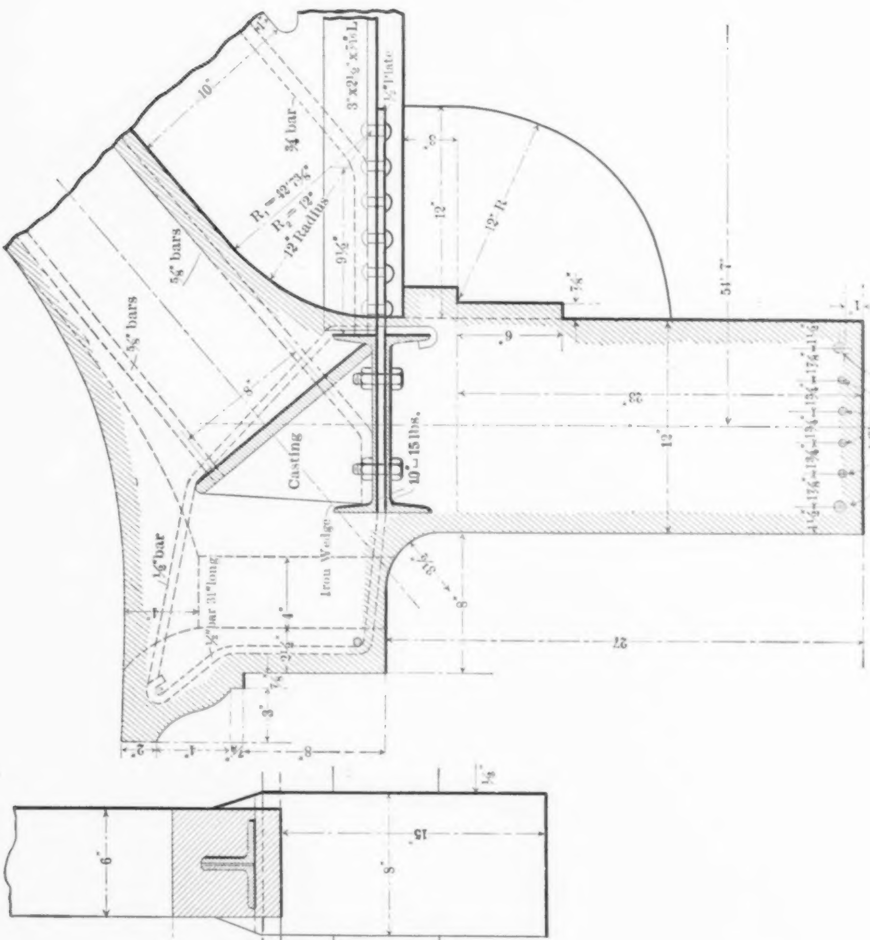


FIG. 10.—DETAIL OF ROOF CONSTRUCTION.

In General.

20. On the front elevation of the building, the concrete mouldings about the arch windows and the cornices are tinted to a dark red color that has a very pleasing effect to the eye in combination with the dull gray of the concrete.

21. The side elevation is almost entirely taken up by windows. The light inside the shop is nearly as good as out of doors.

22. Fig. 13 shows a cross section with crane in place, giving general dimensions of beams, columns, etc.

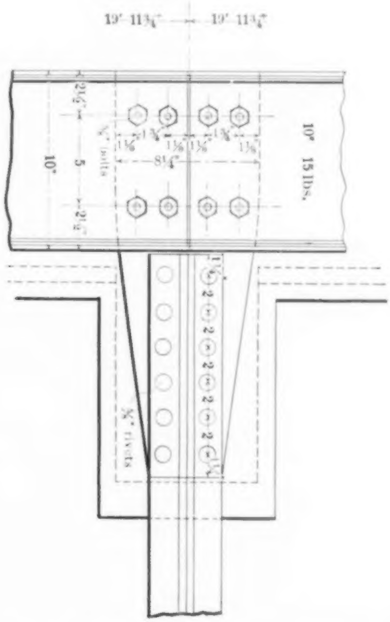


FIG. 10A.—DETAIL OF ROOF CONSTRUCTION PLAN.

Conclusion.

23. The unit cost of this work was approximately 3¼ cents per cubic foot of contents of the building. The building as completed is as near fireproof as is possible in any style of construction to build. As for repairs it will need none. Paint is unnecessary on its surface to protect it from the elements. The insurance rate

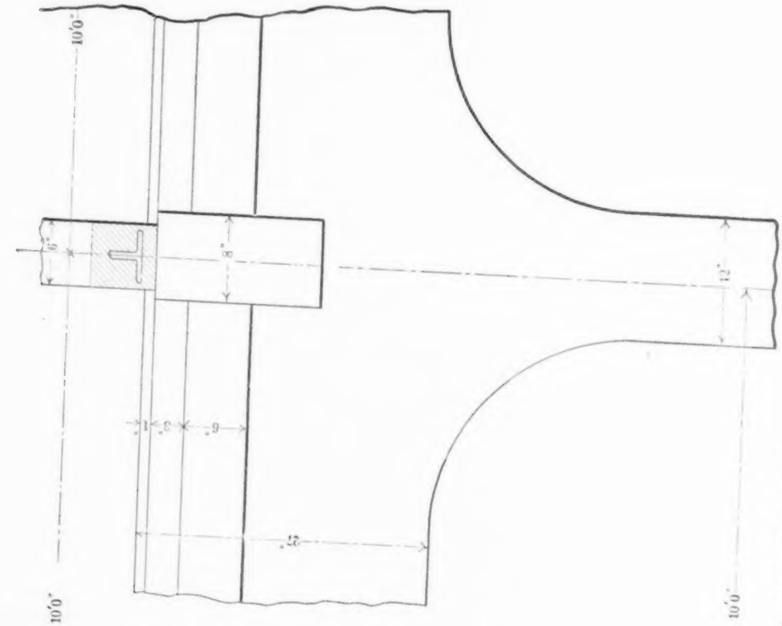
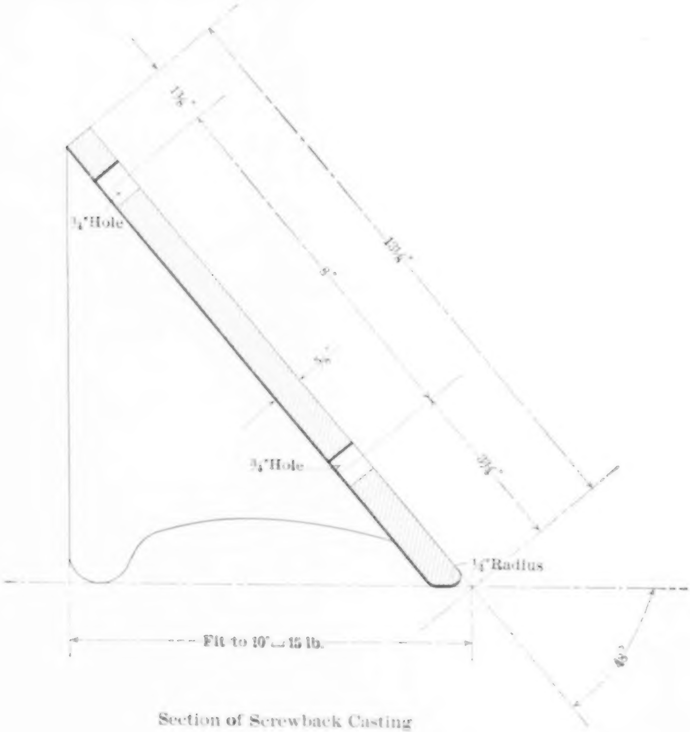


FIG. 10B.—DETAIL OF ROOF BEAM.

made by the Board of Fire Underwriters is 30 cents per hundred for building and contents compared with 50 cents for structural steel machine shops equipped with automatic sprinkler systems,

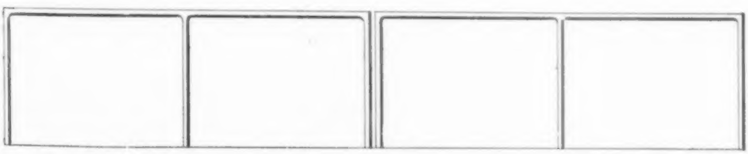


Section of Screwback Casting

FIG. 10C.

and it is considered by them to be the best risk in the Pittsburg district.

24. The works were designed and built by Robert A. Cummings, M. Am. Soc. C. E., of Pittsburg. The writer acted as his



Detail of Roof Construction

FIG. 10D.—SKEWBACK CASTING.

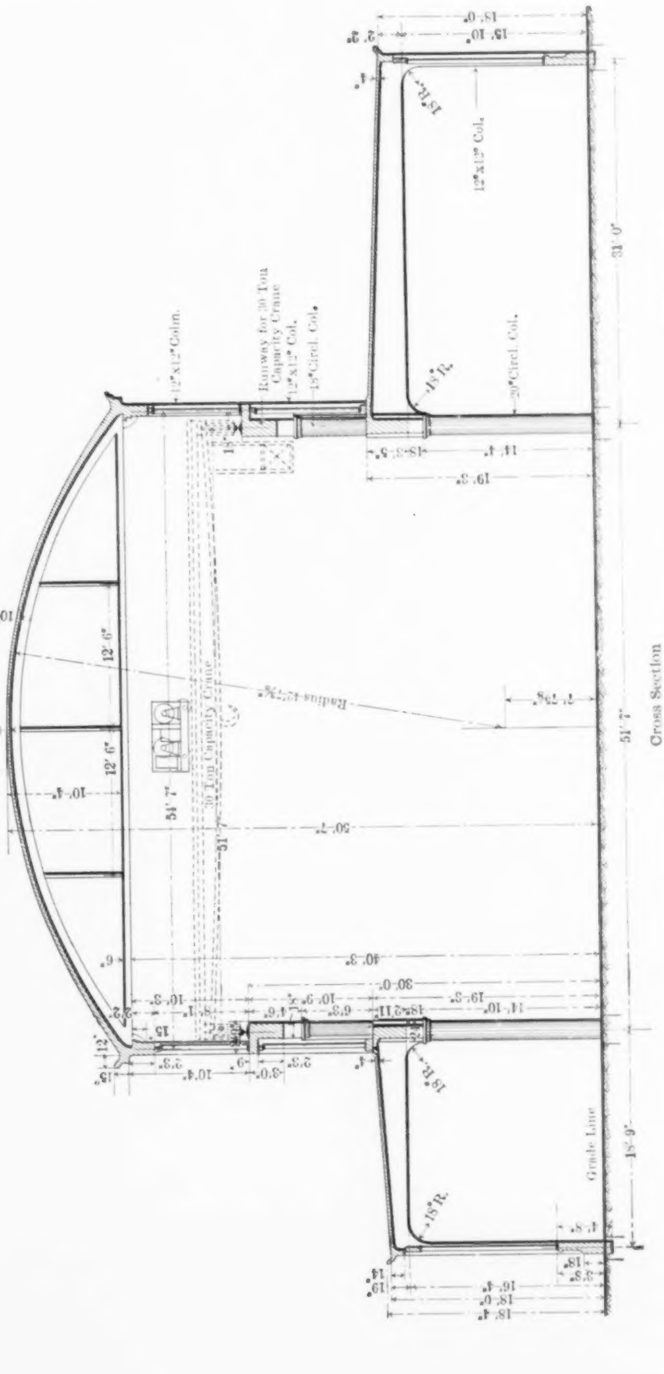


FIG. 11.—TAYLOR & WILSON MACHINE SHOP, MCKEES ROCK, PA. REINFORCED CONCRETE.

assistant in charge of the work during its construction, and is indebted to him for data and information accompanying this description. Much credit is due the principal members of the Taylor & Wilson Manufacturing Company for their foresight and progressiveness.

DISCUSSION.

Mr. Joseph C. Schaeffler.—I have read the paper entitled "Reinforced Concrete Applied to Modern Shop Construction" with a great deal of interest, but I think that it is not general enough in its treatment of the subject. Both the system here described and that patented by Mr. Julius Kahn make use of diagonals which make, theoretically, an excellent reinforcement for concrete beams, but, practically speaking, the superiority of such construction over the beam reinforced with plain rods which have been anchored by turning up the ends or in some other manner, is a matter of some doubt. When Kahn or Cummings bars are employed, the concrete, to obtain good results, must be made of small stone or gravel and placed very wet. Actual practice shows that concrete made in this way and worked rather than tamped into place gives very good results. If large stone were used and tamping resorted to, it is the belief of the writer that the diagonals would be bent down out of position, thereby defeating their theoretical value.

I note from Figs. 6 and 7 that the reinforcement in the arched roof consists of $\frac{5}{8}$-inch round rods. It would be of interest to know whether these rods were made continuous across the entire arch by welding or whether the short lengths were merely lapped to make joints. I should also like to know whether some mechanical bond was used in connection with these rods, or is the adhesion of the concrete to the plain rod considered sufficient. I am of the opinion that the vibration due to the operation of the crane and shafting would seriously affect the adhesion of the concrete to the plain rods, and that in time this would result in a serious weakness. In connection with this, Mr. A. L. Johnson, of the American Society of Civil Engineers, may be quoted as having made the following statement: "In our experience we have had cases of rupture of the adhesion of concrete with plain bars after eight years' use, where the stress in the bars did not ordinarily amount to much, the failure being due entirely to vibrations and shocks."

Referring to paragraph 23, I would like to know whether the unit cost of three and one-quarter cents per cubic foot includes forms. On all concrete work the cost of forms is quite a considerable item, amounting to twenty-five per cent. and over of the entire cost of the placed concrete, and must be included in the statement of final cost. The cost given is somewhat lower than anything else of a similar nature that I know of at present.

I must take exception to the statement made by Mr. Hunting under the title of "Adaptability," where he says that the mixture can be made with equally good results by very efficient mechanical devices or by the use of the most ignorant class of labor. There is no doubt whatever about the superiority of the mechanical concrete mixer over the hand method, both in the matter of final cost of concrete and also in regard to the quality of the mixture. I have seen stuff that could hardly be designated as concrete mixed by hand under the supervision of an intelligent (?) inspector or foreman and lowered into the new subway now being built in Boston. The same applies to about 50 per cent. of concrete which is mixed in this way, and the reason for it is that the only aim of the average foreman is to rush his work as much as possible and thereby make what appears to be a good showing.

I believe that this is the first paper on "Concrete Construction" to be presented at a meeting of the American Society of Mechanical Engineers, and in closing I would like to ask if Mr. Frank B. Gilbreth, a member of this Society, could not be induced to write a paper descriptive of some of the various mills and buildings he has so successfully erected during the past few years.

Mr. H. C. Turner.—While concrete as a building material has been in use for a very long period of time, dating back to at least the ancient Romans, the material known as "Reinforced Concrete" is of comparatively recent origin. It is generally accepted that the first reinforcing of concrete with steel bars or wire was done in France some thirty years ago. From this early work has gradually grown up the scientific combination of the two materials in a standard form of building construction.

Probably the first work done in this country with this material was that by Mr. Ernest L. Ransome in California in the early seventies, and much work was done in that State before any use was made of it in the East for building construction. The large factory of the Pacific Coast Borax Co., in Bayonne, N. J., erected by Mr. E. L. Ransome in 1898, was the first application of rein-

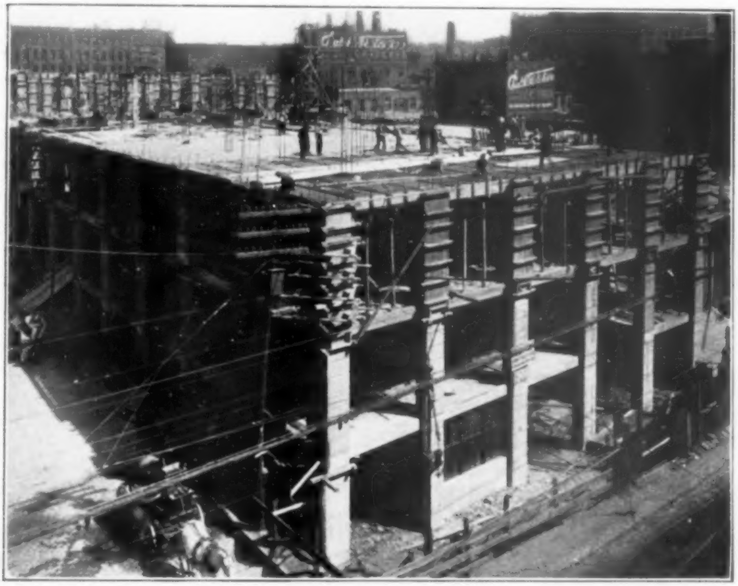


FIG. 12.

FIG. 13.

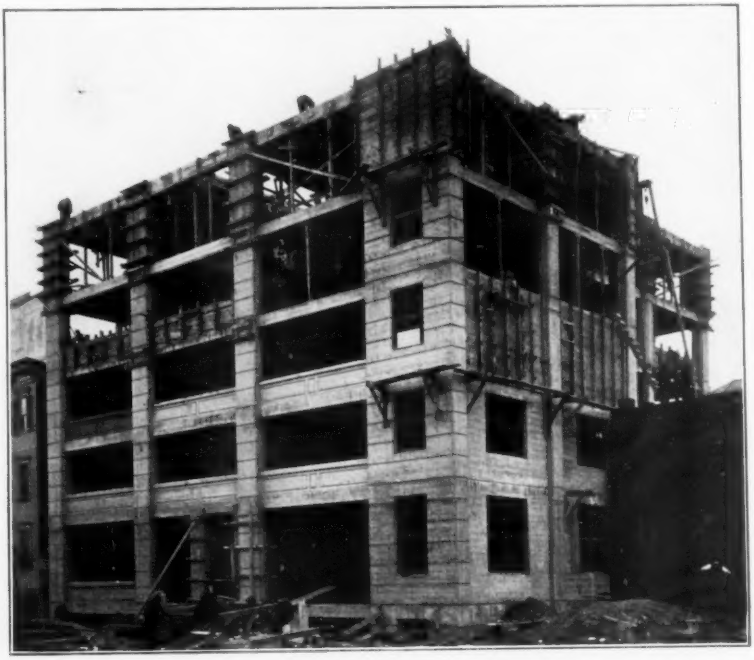


FIG. 14

FIG. 15.

forced concrete to factory or mill construction on a large scale. Dating from this building, the use of reinforced concrete construction has grown very rapidly in all classes of construction work for bridges, sewers, water works, tunnels and buildings. The successful construction of the fifteen story Ingalls Building in Cincinnati of reinforced concrete in 1903 gave great impetus to the construction. Much of the construction of the new Naval Academy in Annapolis is of reinforced concrete, and it has been specified and accepted for the new military academy at West Point. The new building of the United States Shoe and Machinery Co., at Beverly, Mass., and the Foster-Armstrong Plant, at Rochester, N. Y., are entirely of this construction.

It is, of course, of prime importance that buildings to be occupied as shops or factories, and therefore filled with many employees, should be conservatively designed and constructed, and any material to be generally accepted for such work must pass satisfactorily certain prescribed tests for durability, strength and fire resistance. Many tests have been made both in Europe and in this country to develop reliable formula for determining the stresses in the combined material and all designs to-day by reliable companies are carefully and conservatively made. The load tests on full-sized floor sections have demonstrated this fact.

Concrete weathers better than most building stone, and is known to improve in strength and hardness with age. With reference to the effect of vibration in factories, all information obtainable so far is distinctly in its favor. The Bayonne factory of the Pacific Coast Borax Company has been in use about six years, and contains both heavy and rapid running machinery. It is in perfect condition, and the vibration is almost entirely absorbed.

The following letter from the American Book Company, of Cincinnati, speaks for itself:

February 24, 1905.

Ferro Concrete Construction Company,
City.

Gentlemen:—Replying to inquiry made by the Turner Construction Company as to the amount of vibration which we find in our building, we take pleasure in saying there is practically none.

Our printing office is a room having nearly 21,000 feet of floor space and built with a carrying capacity of 350 lbs. to the sq. ft. We are running in it 25 presses, most of them weighing about 13 tons, and have ample space for additional machinery as it is required. There is practically no perceptible vibration when

standing by the press, and no perceptible vibration whatever in the columns or walls. The building is in this respect superior to our expectations, and we have no hesitancy in saying that a building properly constructed of concrete is an ideal one for machinery in motion.

We take pleasure, also, in stating that the work done by your company for us in this building was very satisfactory. We think exceptional care was employed, and the engineering was skillful. We are very glad that we decided upon a concrete structure and upon your company to do the work.

Very truly yours,

(Signed) AMERICAN BOOK COMPANY.
W. B. THALHEIMER,
Managing Director.

The Ketterlinus Litho. Manufacturing Company have recently built an addition to its Philadelphia factory; the old building consists of brick walls, steel columns, girders and beams, and terra cotta arches, with finished floors of maple. Before erecting the new portion the vibration from the heavy presses was very excessive; no clock could be kept running in the building. The addition is constructed with concrete walls, veneered with brick, steel columns, fireproofed with reinforced concrete, reinforced concrete girders, beams and floors, and a finished floor of maple. Since the addition has been built the vibration in old building is largely reduced, but there is still a very noticeable difference between the vibration in the concrete portion and the brick and terra cotta portion of the building.

There are also a large number of factories having light running machinery, but it is generally acknowledged that printing presses are as severe a test as any, and the above instances indicate that reinforced concrete will very largely overcome vibration in mill buildings.

A building material, to be acceptable, must withstand the combined action of fire and water. The Department of Buildings in New York City has conducted a large number of tests on various building materials, and before being accepted for use in fireproof buildings, they must pass satisfactorily a combined fire, water and load test. These tests in recent years have been made very largely by Prof. Ira H. Woolson, of Columbia University, for the Department, and copies of his reports may be obtained by application to him. His report of the Ransome system, outlining the character of the test and the action of fire, water and load on this construction is very complete.*

* A résumé of this report is given in Prof. A. L. Williston's discussion of this paper.

There have been six or more similar tests on the different systems of concrete construction, all really tests on the fire resistance of concrete, their success not depending upon the form of bar used. The fact that concrete can withstand such very severe tests is sufficient proof that it may be conservatively adapted for factory buildings. The heat undoubtedly drives off some of the water of crystallization, thereby reducing the strength of the concrete on the face of the parts exposed to the heat. This loss of strength is very slow, and a heat of 1700 degrees Fahrenheit kept up for four hours will not penetrate the concrete more than about one inch. It is not reasonable to expect any such duration of heat in service conditions. It is also interesting to know that this concrete will again take up this water of crystallization and regain very largely its lost strength. The fire in the Bayonne factory of the Pacific Coast Borax Company which occurred in 1903 did not damage the concrete construction more than $1,000 in a $100,000 building, and there was no damage to the concrete that would obstruct the immediate resumption of manufacture. The Baltimore fire also demonstrated very clearly the fire resistance of concrete.

With reference to cost, it is impossible to give definite figures or to make comparisons that will apply to all localities. In New York and Brooklyn six-story factory buildings of reinforced concrete will not exceed the cost of mill buildings of brick and yellow pine more than five to ten per cent., and under some conditions not as much as five per cent. Reinforced concrete has the added advantage that the height of the building is not limited to six stories by building regulations as is the case in mill construction, and the owner can erect a larger building on his property, and, therefore, have a better investment.

In concrete buildings, construction is of equal if not greater importance than design, and all work should be done under experienced and competent supervision. The cement, sand and stone should all be carefully inspected and proportioned to give the best concrete. The cement should be regularly tested; the concrete should be machine mixed, and the steel should be secured in position before concreting. Tests should be made regularly on the concrete by means of cubes made from each day's work, and load tests made on the completed floors.

With reasonable care, excellent buildings can be constructed, and the advantages of reinforced concrete are so great that it

should receive the careful investigation of every engineer before deciding upon the design of factory or mill buildings.

There are under construction this year in New York City the following buildings of concrete:

Brooklyn: Robert Gair Company factory, 8 stories and basement, 100 x 200 in plan.

Hoole Machine & Engraving Works building, 4 stories and basement, 25 x 63.

Gretsch factory, 6 stories and basement, floor areas, 9,000 sq. ft.

C. Kenyon & Co. factory, 6 stories and basement, 80 x 145.

Bush Co. Model factory, 6 stories and basement, 75 x 600.

Bush Co. Warehouse, 8 stories and basement, 150 x 300.

Hanan Shoe Co. factory, 5 stories and basement, 120 x 200.

L. I. City: McClure & Co. factory, 6 stories and basement, approximately 100 x 260.

N. Y. City: Schirmer factory, 4 stories and basement, 50 x 100.

Also J. B. King & Co. factory at New Brighton, S. I., 4 stories and basement, 45 x 60 in plan.

Mr. H. F. J. Porter.—The tendency in building construction nowadays, as exemplified in the paper under discussion, is manifestly in the direction of fireproof construction for the obvious purpose of the absolute prevention of fire, and resulting in the protection of lives and property from such a casualty. It will, however, be many years before all factories will be fireproof, and meantime it becomes necessary to consider the best means for adoption to protect the lives and property housed in such factories as now exist and will continue to exist for a long time to come. The laws of the land and of the community in which a factory is situated require the fulfilment of certain obligations to insure such protection, but the moral responsibility which rests upon the employer of labor who assembles many people to work for him should require him to do many things for their protection which it is evident cannot be embodied in the law. And it is evidently to his interest that every consideration should be given to effect their prompt escape from the building in case of fire, for it is the first duty of a fire department to save lives before attempting to save property, and in a case where lives are in jeopardy, attention must first be directed to their relief, and meantime much property may be destroyed.

Consequently, it is desirable to arrange so that the means of promptly preserving lives are efficient, so that as soon as possible after a fire starts escape from it may be effected and attention be given to putting it out. In this connection experience shows that there are more casualties resulting from panic occurring during fire than from the fire itself, and also that panic frequently occurs simply on the alarm of fire when actually no fire exists, and on other occasions when there is a similar suggestion of danger. It therefore devolves upon the employer of labor to take such preliminary precautions that not only will his employees be able to obtain a prompt and easy exit from the building in case of fire, but also be protected from the results of panic in such an emergency or of panic due to any cause whatever.

It would be well to consider the methods now in vogue for protecting the lives of employees in the ordinary factory, of which there are hundreds in this and every large city of the country. In all of these places we find fire buckets; in many of them stand pipes with hose attached, and in some of them sprinkler systems, but the factories so provided are few which have assigned the duty to certain specified individuals to use the buckets or the hose in case of fire or to turn off the sprinkler system after the fire is out; and what is anybody's is nobody's business, especially under stress of excitement. All of these appliances can, however, become effective by the development of a regular fire corps.

The fire escapes which are supplied to buildings generally are of such construction as to be a more prolific source of, than escape from, accident. They generally consist of an iron balcony on the outside of the building at each floor, and these balconies are connected by means of ladders, the lowest balcony having its ladder detached and hung up on the building so as to prevent people from entering the latter, which they might if it was permanently placed on the ground.

These ladders vary in position from the verticle to an angle of 50 or 60 degrees with the horizontal, and in order to descend, people have to turn around and go down backwards. In all cases of emergency, when both women and men have to descend by such means, and especially in winter weather, when the iron is cold and perhaps covered with snow or ice, these so-called escapes are prolific sources of accident, especially at the lowest balcony, where if the ladder is a long one, it is correspondingly heavy, and the combined strength of several men is necessary to handle it.

These men are not necessarily the first to reach the spot where it is located, and they cannot well get there after others have crowded down ahead of them. There results on such occasions a jam, accompanied by a crowding off of the balcony those who are near the opening. Many of these escapes can be improved by making slight changes in them.

It has been stated that the mortality from factory fires is very low. It is doubtful, however, if there are reliable statistics to show the results of such casualties.

Since the announcement on your program that there would be a discussion of this subject, I have taken the trouble to collect clippings on the subject of factory fires from the press of several large cities, and I find that there are a number of casualties of this kind occurring every day, involving the lives of many employees. The latter are carried from the factory to their homes or to hospitals and their cases are subsequently lost sight of.

The fire department of this city advises me that almost daily they have a fire in a factory where they have to save lives before they can direct their attention to extinguishing flames.

In a factory where I was engaged a couple of years ago, finding the conditions such that in case of a fire either within the building or in adjacent buildings it would be necessary to promptly empty the building of its occupants, I established a fire drill which, by occasional repetition, reduced the time of exit from more than seven to less than three minutes, a saving of time which in the case of a rapidly burning fire or one involving considerable smoke would undoubtedly have been the source of preserving many lives. Since that time I have introduced similar drills into other factories, and in every case with remarkably desirable results. Whereas at first such a drill, owing to its association with danger, is usually accompanied by the fainting of some of the girl employees, later a habit of prompt and rapid dismissal is acquired, and all become possessed with a feeling of security which largely counteracts the tendency towards a panic.

In one of these, a clothing factory employing nearly 1,000 hands, both men and women, on the occasion of an altercation between men in an adjoining railroad yard, one of the employees called out "fight." Mistaking the call to be "fire," a panic occurred during which the factory was emptied with many accidents, one man leaping from a second story fire escape and breaking his leg. Since the fire drill has been introduced, there

is no tendency towards a panic. Everyone knows there is no danger at least until the fire signal is given, and that then they will be properly taken care of.

In another factory, where 1,000 girls were engaged in making shirt waists, the appearance of a rat occasioned a stampede which did not end until many girls were injured. There have been similar occurrences since the drill has been introduced, but the girls have gone quickly to the nearest exit without excitement.

Some clippings which I have here show that panics occur where there is no danger. One from the New York "Times" speaks in its headlines of fifty girls becoming hysterical and several fainting in a shirt factory where a newspaper blazed up from a spark from an electric motor, and that many women were trodden down in a senseless panic by the men, who were the more terrified. Another clipping from the same paper speaks of the employees of a large manufacturing plant becoming frightened by the smoke entering the windows from a burning tar pot in the street, and that in their rush for the exits a dozen were so injured as to require the attention of surgeons.

Other clippings show that danger is occasioned by the carelessness of some of the employees themselves. The New York "Herald" of November eleventh speaks of 400 girls who were left locked in on the fifth story of a burning underwear factory by the watchman, who went to the corner to send in the alarm, and that many leaped into nets held by the firemen and others were taken down on the fire ladders. Still another account tells of a fire in a feather factory, where the girl employees went down the fire escapes until they reached the last balcony, where they could not handle the heavy portable ladder, and then on account of suffocation by smoke were carried, fainting, to the street.

I have here, however, other clippings showing that fire drills have in similar instances saved lives. In a factory in Chicago, for instance, scores of employees on the signal of fire formed and marched from a building which was shortly afterward burned to the ground. An editorial from the New York "Sun" speaks of the efficiency of a fire drill in a hospital and a school where large numbers of children were in danger of destruction by fire, but quietly marched from the buildings; whereas, in two factories the adult employees were crazed with fright by an imaginary danger and injured one another and destroyed much property in their wild flight for safety.

In many factories which I have examined at the request of the proprietors to satisfy them that they were doing all they could for the safety of their employees, I have failed to find any which in one way or another had not introduced some obstruction to the availability of their fire escapes.

Many factories give no thought to the subject after the fire escapes and fire buckets are supplied, and on one occasion the question to the superintendent, what he would do in case a fire occurred right then was met by the amazing reply that he would think of some way to get his employees out. When pressed to think then, while he had plenty of time, of a way to do so, he was utterly unable to say how under conditions as they existed he would have gotten them out.

It is only by going through the maneuvers which would occur in case of an actual fire that the efficiency of the appliances and escapes can be tested. A fire drill will do this, and, once introduced, its desirability is too evident to allow it to be discontinued.

Mr. George Hill.—Paragraphs 1, 3, 4, 5, 6, 7, 8 are heartily endorsed. The paper as a whole is to be commended for clearly stating the points of interest to mechanical engineers and for omitting those matters relating particularly to construction, which are more appropriately the province of civil engineering.

It is especially noteworthy and commendable that the author has given in paragraph 23 the cost of the building and the insurance rate obtained, because these facts present the commercial aspect. It is to be hoped that the author will, in the conclusion of the discussion, state the costs separately for each lean-to and the center aisle (showing the influence of span and excessive height on the cost), and also state what kind of floor was provided in the cost stated.

The writer presents photographs (Figs. 16 to 22) illustrating two plants, completed by him four and five years ago, for the purpose of illustrating certain points he wishes to discuss.

Costs: The one-story building, of which Figs. 16, 17 and 18 are interior views, are made with cast-iron columns, reinforced concrete roofs covered with a tar and gravel surface, each column supporting 400 square feet: Side walls of expanded metal lath, plaster and portland cement 2 inches thick; floors 3 inches by 3 inches yellow pine on 6 inches of cinder concrete in which sleepers were imbedded, each piece of flooring

Fig 16.

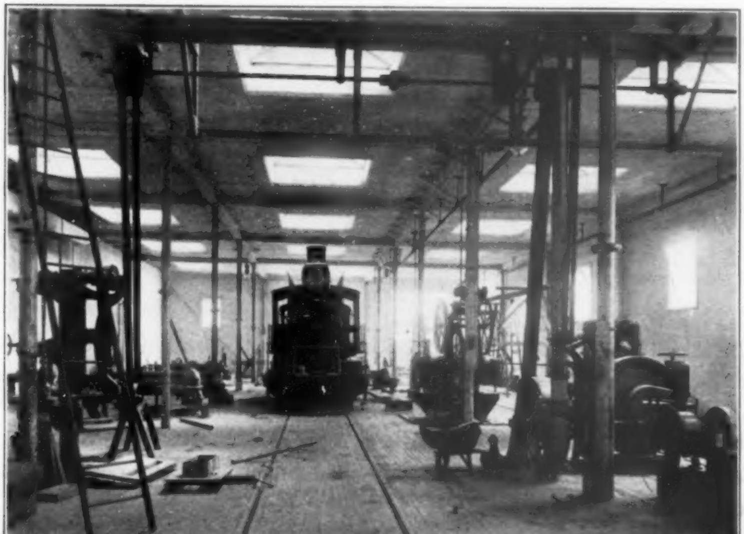


Fig. 17.

surface nailed. The skylights had galvanized iron frames covered with translucent fabric in place of glass. The buildings cost about 4.1 cents per cubic foot, or 85 cents per square foot, made up as follows:

Floors 25 cents per square foot: Roof (including columns and centering), 50 cents: Skylights, 28 cents per square foot of horizontal opening between curbs: Side walls, 20 cents per square foot of gross area.

The erecting shop (Fig. 19) had two lean-tos, each 60 feet wide,

FIG. 18.

and a central portion 60 feet wide, 50 feet to the underside of the trusses. The ends of the central part were of corrugated iron hung to a steel frame, the roof 3 inches yellow pine plank, covered with tarred felt. This building cost, as a whole, 3.4 cents per cubic foot, the lean-tos costing 4.1 cents, and the center aisle costing 3 cents.

The engine house (Figs. 20, 21) cost about $2 per square foot of ground covered, but there was a finished basement in the engine room portion and an exceptionally high ceiling and heavy coal bunkers in the boiler room part. The storehouse (Fig. 22)

Fig. 19.

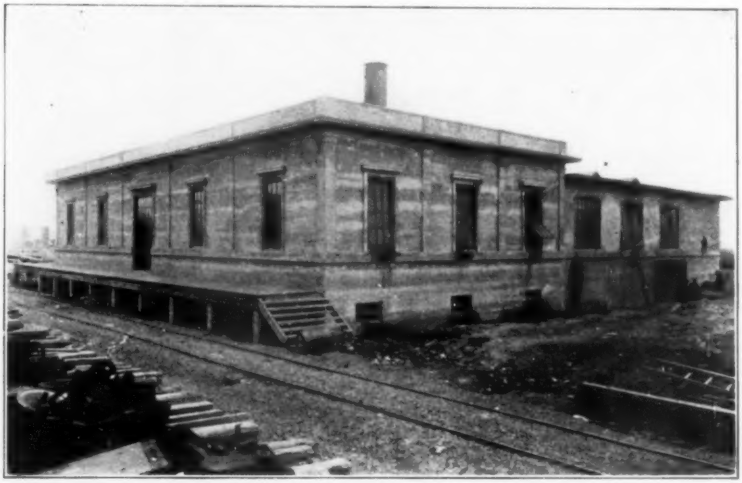


FIG. 20.

FIG. 21.

cost about $1 per square foot of floor area obtained. A building with concrete floors laid on a cinder fill and a roof constructed of heavy timber and 3-inch plank, nearly flat and presenting the same general outlines as the reinforced concrete roofs shown, costs about 42 cents per square foot with simple skylights; 48 cents with the modified sawtooth skylight.

Foundations: The cost of foundations depends on the character of the ground, on the weight of the buildings and on the method of construction employed. Reinforced concrete is es-

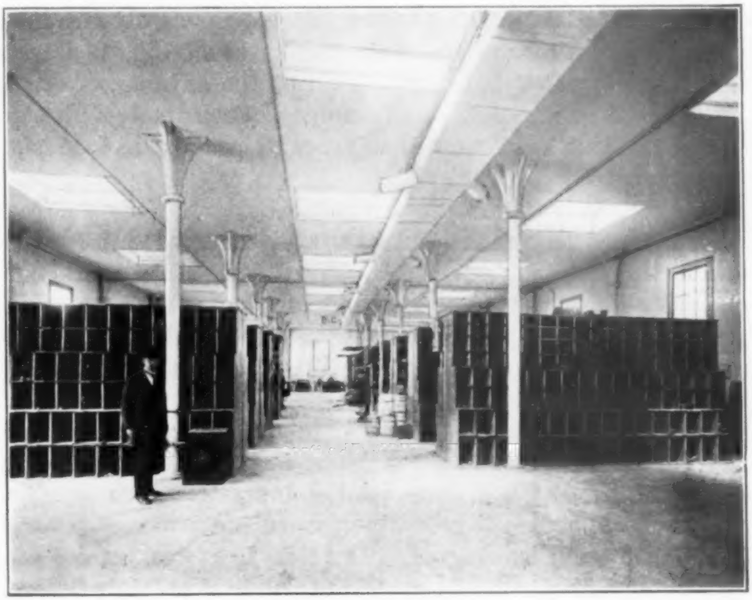


FIG. 22.

pecially valuable in that it is in a measure quite elastic, so that unequal settlements do not injure the roofs. On the other hand, concrete side walls 4 inches or more in thickness will almost certainly develop objectionable or unsightly cracks unless especial care and appreciable expense are expended on their foundations; for this reason the writer advocates 2-inch curtain walls of wire lath and plaster or light brick walls, either being less expensive and more readily kept in order.

Lighting: Fig. 1 of the paper shows a well lit building depending on side light entirely. This form of lighting demands special

treatment of the window surfaces exposed directly to the sun, since the direct rays of the sun are unbearable during a large part of the year.

Overhead lighting, especially for buildings more than 50 feet wide, is a necessity, and the familiar forms of skylights, either isolated or of sawtooth form, are known to all. The writer shows (Figs. 16 and 17) the application of translucent fabric in place of glass to known forms that gives at all times an almost perfect illumination. It is not necessary to screen these lights, and the objectionable high lights, requiring a constant adjustment of the retina and a consequent tiring of the eyes, are absent. The buildings shown toward the right hand rear of Fig. 19 have a different form of skylight, giving all of the advantages of the sawtooth with none of its disadvantages. Such roofs, with skylights, should cost not more than 55 cents per square foot complete.

The skylight is made with two vertical sides, ceiled; the front is made nearly vertical and furnished with a hinged sash for ventilation, the top is a single slope covered with fabric and admits practically all of the light—some coming from the front, which should face the north. In a building 20 feet high excellent lighting will be obtained if there is 40 square feet of skylight opening in each 400 square feet of roof.

Layout: The writer is a believer in and advocate of the one-story factory wherever it is possible to employ it. Properly designed, it will cost less to build, heat and operate. If the machinery and departments are properly grouped and arranged the cost of production will be lessened. The risks from fire, both of partial or total losses, are much less and the distances partially finished products are transported need not be increased.

Durability: Reinforced concrete may be made as durable as commercial conditions demand. It is desirable that there should be some expression of opinion on the part of the members of the Society as to how long these factory buildings should be expected to last. There are cases where only a year or two would be enough, other cases where 5 years would do, but are there any cases where a wise business policy would permit a set of buildings to stand unchanged for say 30 years, unless the locality in which they stand had ceased to develop? There may be an affirmative answer from the cotton mills which are 30 or more years old and certainly alive and progressive, but how about other industries?

The translucent fabric mentioned is made by dipping fine mosquito wire netting into a linseed oil mixture, resulting in a slightly sticky covering, about 3-32 of an inch in thickness; it is readily ignited by means of a match, and if the sheet is held vertically will burn until all of the surface is charred. Applied to a skylight having a wooden frame, it was tested by using as kindling about one bushel of pine shavings, kindling and light wood until the fabric had apparently been thoroughly ignited. The wood was then swept off the skylight and the fire died out in about three minutes, extending only very slightly beyond the area that was ignited. A second test was made by opening the ventilators at both ends of the skylight, smashing a hole nearly one foot square through the fabric and igniting it with shavings as before. In this case the flames had access to both the under and upper side of the skylight and a good draft existed. When thoroughly ignited the kindlings were again removed and the fire died out in four minutes, the area affected being only slightly greater than in the first test. The same skylight was used for both tests—it was repaired in two and one-half hours of time by one man by the application of new fabric and a little paint. The fabric should be applied with a slope of 5 in 12 preferably of the modified sawtooth form shown. It can be cleaned, is not easily ruptured and is very easily repaired.

The writer is a strong believer in the employment of flat slab roofs, and in the buildings illustrated in the photographs followed his belief as far as he was permitted by his clients. The advantages are the minimum of cube of building required; the avoidance of dead air spaces which accumulate dust and dirt; facility in the erection of all overhead appliances; slow runoff of rain water rendering its collection and saving easy; the retention of the snow falling on the roof which acts as an insulating cover in the winter time; the decreasing of the actual cost of the building, although slightly increasing the cost per cubic foot of building.

All of the buildings illustrated in the photographs were heated by the hot blast method, in some cases the hot air being carried through tunnels below the floor line and discharged into the room through branch pipes 8 or 9 feet high, and in other cases through pipes suspended from the ceiling. This latter method is somewhat less expensive than the former unless the tunnels are a necessity for other purposes. In all cases the supply of fresh

air to the fan was arranged so that a portion or all of it could be obtained from the shop, so that in extreme weather the air in the shop was circulated over and over again, leakage and the opening of doors keeping it fresh, while in moderate weather fresh air from the outside was driven in.

In general, the cubic feet of air space per employee in a shop is so very large that there is no objection to circulating the shop air, while there is a very material saving to be effected by so doing. The reinforced concrete buildings, as shown, are very warm, as the concrete is a poor conductor; the principal heat losses occur through cracks around the door and window openings and glass radiation. The translucent fabric radiates less heat than the glass. The 2-inch plaster walls are probably as efficient as an 8-inch brick wall. The writer has made no experiments to determine this, but finds that heat computations based on this assumption provide a sufficiency of heat units to render the buildings comfortable.

In factory construction, so far as fire risks are concerned, we are in the habit of regarding a fire as an unavoidable infliction which must sooner or later visit every factory. It might almost be said that we consider it necessary to build factories which shall contain a sufficient amount of inflammable material to render a fire therein serious. As a consequence, very efficient systems have been developed for fighting fire—with the advent of a reinforced concrete construction this should no longer be necessary. If factories generally were built along the lines illustrated in Figs. 16 and 17, no fire protection would be required other than a few chemical extinguishers to be used in the event of a contents fire, no insurance would have to be paid, employees would be absolutely free from danger. Should multiple story buildings be necessary, it is only necessary to make the floors of reinforced concrete slabs, the side walls either of the curtain construction shown or of brick, and to inclose all openings through the floors with tight shafts to secure an almost equal degree of safety. In a building so constructed the employees will be safer if they remain on their respective floors in the event of a contents fire than they would be in attempting to escape by passing down the stairs or down the fire escapes, even the best of stairs being dangerous when used by those excited by the fear of death.

Referring to the matter of costs: A statement of cost per cubic foot of building is very apt to be misleading since a relatively

expensive construction may by reason of excessive height produce a lower cost than was obtained in a similar building erected in a cheaper manner but with less height. The writer, therefore, prefers a statement of cost per square foot of floor area obtained. The cost must of necessity vary materially with the locality, since different localities afford different opportunities for procuring the aggregate constituting the bulk of the concrete. It has been the writer's practice to allow an average of $1\frac{1}{4}$ barrels of cement per cubic yard of concrete, the actual proportions employed varying from one-third of a barrel to nearly two barrels per cubic yard. This fixes one element of the cost. If the constructive design is commercial, and if the steel is disposed in the proper manner without attempting to secure the maximum amount of work therefrom, that is, if probably five per cent. more steel is used than is really necessary, the placing and the mixing of the concrete may be performed by the most unintelligent of laborers with perfect safety. These men can be driven so that the labor cost per cubic yard of concrete for spreading, mixing, depositing and ramming may be brought to eighty cents per cubic yard. If engineering stunts are attempted, such for example as the arched roof shown, the greatest possible care must be given to every detail of the work, with a consequent increase in expense, and this is not good business.

Prof. A. L. Williston.—I wish to refer to the tests that Mr. Turner has spoken of. They were made by Prof. Ira H. Woolson, of Columbia University, in coöperation with the Bureau of Buildings of New York City and the Turner Construction Company.

The purpose of the tests was to discover the effects of long-continued and intense heat, such as occurs in a general conflagration on concrete construction; and also the effect of the sudden application of water on concrete when it has been thoroughly heated.

A small building of reinforced concrete about 18 feet long, 13 feet wide, and 11 feet high, with a top which was so constructed that it could be used for floor test as well as for a roof, was erected for the purpose of the tests, and was subjected to fire and water tests and load tests. Two firing doors were placed in the east wall of the building and draft openings at the bottom and smoke openings at the top were provided, so that an intensely hot fire could be maintained in all parts of the interior chamber. The fuel used was wood, which was fired on a roughly constructed

grate over the entire area of the chamber in such a way that there was a strong and continuous draft passing up through all parts of the fire.

A dead load of 150 pounds per square foot was placed on the floor above the chamber before the fire was started, and this load was maintained during the fire test. Careful measurements were made of the deflections during the fire test, and both before and after the fire was started. After the building had cooled down a load of 600 pounds per square foot was applied and measurements of deflection were again made.

The fire test lasted a trifle over four hours, during which time a mean temperature of about 1,700 degrees Fahr. was maintained. At the end of the fire test, while the building was still red hot, a $1\frac{1}{8}$-inch stream of water was played on the inside of the building, directed onto the ceiling and the different sides of the building for about five minutes. The floor above was then flooded for five minutes. And for five minutes more a jet of water was played on the lower side of the ceiling. The stream of water was applied at short range and under a pressure of 60 pounds.

During the fire it was noticed, in a few places, that the concrete scaled off slowly, with sharp little explosions, caused by the sudden generation of steam from the moisture in the walls. The walls were full of moisture, as was evidenced by the fact that soon after the test started the water began to sweat through onto the outside. This greatly increased until the water ran freely down the sides. It continued for about two and a half hours, and the amount of water was surprisingly large.

Observations made just before the application of the water showed that the blowing off of the concrete had done no material damage to the building. The columns and a few spots on the ceiling were chipped off in patches a foot or more long and a little over an inch deep. There were also numerous patches blown off from the sidewalls, but the girders appeared to be in nearly perfect condition. When the water was applied more of the concrete chipped off in places. The floor slab was pitted to the depth of $\frac{1}{4}$ inch, and a few cracks which appeared to be superficial were noticed. The concrete on the bottom of one of the girders at the middle was knocked off and exposed the metal rods for a space of several feet, but this was the only place where the metal was exposed, and none was exposed before the water was applied. With the exception of the few slight defects which I have men-

tioned, the whole building seemed to be in excellent condition at the end of the fire and water test.

After the building was entirely cooled down a load of 600 pounds per square foot was applied over the entire floor to ascertain whether the structure had been weakened by this fire test or not. The cracks above referred to were not enlarged by the application of this load, and the deflection caused by it was practically no greater than would have occurred had the load been applied before the fire test. The maximum warping due to the fire, while the fire was in progress, was $\frac{3}{4}$ inch, the greater part of this being caused by the expansion of the walls. The intense heat lifted up the side walls as well as slightly arched the floor. After the building had cooled down again the floor returned to nearly its original position, the maximum permanent change in any part of the floor being but $\frac{1}{2}$ inch. The maximum effect of the 600-pound load over each square foot of floor was also less than $\frac{1}{2}$ inch.

I will not go into any greater details regarding these tests because the report of them is already in print, but I felt that, in connection with the paper which has just been presented, the Society would be interested in these few facts which I have presented.

Mr. Geo. Hill.—May I say one word: The cost stated in the paper, I believe, and the cost stated by myself I know are not per cubic foot of concrete, but per cubic foot of finished building—a very different matter. I would say, in regard to roof slabs, that my belief is that our factory buildings should be one story in height. The flat slab is a most excellent thing to hang things from, and is protected from the weather on the outside with a tar or gravel or felt roof; and it is good.

Mr. Fred. W. Taylor.—I wish to particularly thank Mr. Hunting for the trouble that he has gone to in presenting the paper. It is impossible to get too much exact information. That is what we want, for it is only from exact and full knowledge of facts that we are able to formulate our theories and laws; and while all that he has given us is very satisfactory, there are, perhaps, a few points which would make his data still more valuable.

As to this cost of $3\frac{1}{4}$ cents per cubic foot, it would be highly desirable to know accurately the elements from which this cost was made up, namely: The price of ordinary day labor as he used it; the price of skilled labor, and the various kinds of skilled labor used; the total amount of skilled labor to be used on the job, and the total amount of ordinary cheap labor; the cost of all the raw

materials on the ground ready for use, and, as far as it is professionally right that he should do so, also the the cost of his forms —I mean the cost of each of the various types of forms which he used.

Some of these forms seem to be remarkably good; others he has not illustrated at all. But if Mr. Hunting has the data, and it is professionally proper that he should give the data to the Society, I think that the addition of that information would greatly supplement the cost side of his paper, and that, after all, is a very important element.

In addition, it is the composition of the aggregate: Not only the amount of cement used, but the size of the broken stone, the quality of the sand, the nature of the cement—I assume it was used extremely wet, but if it was not used wet, then the degree of moisture used.

All those elements will add very greatly to the value of Mr. Hunting's paper, and as he has shown such thoroughness in presenting the paper I trust that he may be able and willing to give us the additional cost elements in the data.*

Mr. E. N. Hunting (in reply to Mr. Schaeffler).—In the application of the Cummings System referred to, I would say that from a practical standpoint it is absolutely impossible to bend the diagonal shear bars down in the form after being set in place. These bars run from ½ inch to 1 inch in diameter, and are rigidly held in place by means of the Cummings Chair Support—a device made up of sheet metal—that clamps around the bars and holds them away from the form.

The unit cost of 3¼ cents per cubic foot of building contents includes the cost of all forms, concrete and steel—in fact, all the structural work complete.

I question the soundness and accuracy of the quoted statement of Mr. A. L. Johnson regarding the failures due to the use of plain bars under vibrating load. Recent investigators have drawn quite different conclusions in this matter.

In reply to Mr. Hill.—The building was provided with a 3-inch oak floor laid on 6 x 6 oak sleepers.

In reply to Mr. Taylor.—The following items may be considered as those used in making up the cost price given in the paper:

Day labor, $1.50—$1.60, ten hours.

Carpenters, $3.50— —, nine hours.

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* Author's closure under the Rules.

About 40 laborers and 20 carpenters employed.

Gas-engine power used—natural gas costing 21 cents per thousand cubic feet.

Cement used was a standard brand of Portland, costing $1.28 per barrel delivered on the work.

River sand and river gravel was used in the concrete, costing 65 cents per ton on the work.

Gravel used was 1-inch ring.

Mixture of concrete used was 1—1½—3 for columns 1—2—4 for the rest of the building.

The mixture was used fairly wet and in a plastic state of sufficient consistency to hold the gravel from settling to the bottom.

No. 1103.*

BEARINGS.

Locomotive Bearings.

Mr. Geo. R. Henderson.—In locomotive practice the gauge of the track, and consequently the distance between hubs of driving and truck wheels, limits the extreme length possible over bearings. The link motion, cylinders, etc., prevent drawing the frames closely together; consequently, the distance from the center of pressure on bearing (corresponding to the location of the spring rigging) to the outside edge of box has often been less than half the desired length of bearing. Under these circumstances the practice of increasing the length by extending the inner side has at times been resorted to. This reduces the average unit pressure, but imposes an eccentric load on the journal.

For instance, a driving journal may have been originally or in a similarly designed engine 10 inches in length, and by adding 2 inches to the inside a box 12 inches long is obtained, the spring saddle maintaining its original position of 5 inches from the outer end of the bearing. It has often been found that such an arrangement gave trouble by heating or uneven wear, and that better results were obtained by shortening the box so that the center of the load would coincide with the center of the length of journal. The average unit load was thereby increased, but a concentration of load was avoided. This can be explained by referring to Fig. 1. In the upper view, the load P is applied centrally as to the length l, and the unit pressure is

$$p = \frac{P}{dl} \quad \dots\dots\dots \quad (1)$$

d being the diameter, and this pressure p will be uniform throughout the length l. If now we add to the inside of the bearing, as shown in the lower view, so that the load P is away from the center

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* Presented at the New York meeting, December, 1905, of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

of the new length by the distance x, the line ab being the center line of the bearing, the unit load or pressure at the edges of the bearing will be

$$p' = \frac{P}{dl} \pm \frac{Px}{\frac{1}{6}dl^2} = \frac{P}{dl}\left[1 \pm \frac{6x}{l}\right] \quad . \quad . \quad . \quad . \quad . \quad (2)$$

the positive sign referring to the edge nearest to P and the negative to the farthest edge.

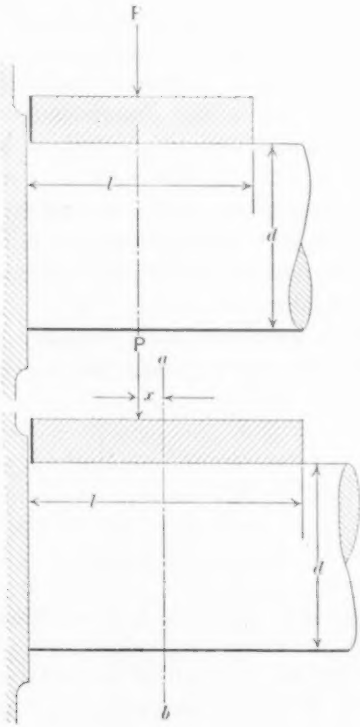


Fig. 1.

As an example, let us take a case in which the load $P = 16{,}000$ pounds, the diameter $d = 8$ inches and the length $l = 10$ inches. If the load be central, equation one gives us for the uniformly distributed pressure 200 pounds per square inch, thus:

$$p = \frac{16{,}000}{8 \times 10} = 200.$$

If now we add 2 inches to the inside of the box, leaving the application of the load as before, we have $l = 12$ and $x = 1$, and from equation 2 we obtain

$$p' = \frac{16,000}{8 \times 12}\left[1 \pm \frac{6}{12}\right] = 166\ (1 \pm \tfrac{1}{2}) = 249 \text{ and } 83,$$

the larger value being the unit pressure at the outside edge, and the lower value the unit pressure at the inside edge of the bearing. While the average pressure is only 166 pounds per square inch, the concentration increases the maximum pressure to 25 per cent. more than with with a 10-inch journal and with the central load.

Of course we do not mean to indicate that ordinarily an engineer would wantonly place a load eccentrically on a bearing, but under the conditions quoted it is often difficult to reduce the unit load to the desired value, especially as it is a serious proceeding to increase the diameter of the axle over what is actually needed, and, besides, an increase in diameter augments the rubbing speed and ordinarily overcomes the benefit of reduced pressure, except as to forcing out the lubricant.

As outlined above, however, it is generally preferable to get along with an increased unit pressure than to create an eccentric loading.

Bearing Metals. Notes Regarding Methods of Testing and Qualities.

Laboratory Tests.

Prof. R. C. Carpenter.—In my opinion there is no method of making a laboratory test of bearing metals which is likely to bring out all the qualities which are required for satisfactory work under all conditions. The Thurston oil-testing machine, which is well known to most members, gives the co-efficient of friction with substantial accuracy. In that machine the load is applied on both top and bottom bearings, which exaggerates the heating effects that usually occur in practice, for in practical cases the load is carried by one bearing alone. This has the advantage of bringing out practical defects in a much shorter time than would usually occur in practice, which is desirable in a short time test. The results obtained with the machine depend more upon the lubricant used than upon the nature and character of the bearing metal, and there

is no definite standard of lubrication with which the methods employed on the machine can be compared. It is a well-known fact that bearing metals under one condition of lubrication may be highly successful and fail entirely under another condition of lubrication.

To overcome these defects I have, in a number of instances, tried a method of testing in which no lubricant should be employed, the relative results being expressed by considering the number of turns required to reach a given temperature for a given pressure per square inch. This method has not been carried far enough to make me at all certain of its value or whether or not it would give comparative results of any practical value.

A method which has seemed most valuable for laboratory results has been that of supplying a limited and known quantity of lubricant and noting the temperatures and coefficient of friction corresponding to different pressures for a given travel of the surface of the journal. In applying a test of this kind the bearing metal will seize the journal at some temperature and pressure, and that has been considered the limiting point which in a measure determines the comparative value of a given bearing metal. This result is highly artificial, as there is no means of definitely applying with certainty a known amount of lubricant to the face of the journal. For the above reasons, the laboratory tests are often misleading.

To determine the value of a bearing metal, it should be tested under very arduous practical conditions and with varying degrees of lubrication for a long period of time. The coefficient of friction is generally a function of the lubrication rather than of the character of the bearing metal, although it is certainly affected by the quality of the metal in some degree.

Desired Qualities.

The qualities which a bearing metal should have in order to be satisfactory are quite varied in nature, and in some respects somewhat contradictory. The bearing metal should first of all be one that has considerable adhesion for a lubricant and is readily wetted by it. It should also be softer than the shaft which it supports, so that in case of lack of lubrication, or in case hard gritty material gets in the bearing, the bearing material would be injured rather than the journal. It should be hard enough, however, to retain its

shape under any conditions of pressure or temperature which are likely to be imposed upon it by actual use. The melting temperature of the bearing metal should be less than that of the journal which it supports, but should, at the same time, be readily melted by changes in temperature which occur in practice. The bearing metal when melted should not possess the property of adhering or welding fast to the journal.

Soft Metals.

For many purposes where the pressures are low and temperature not likely to get high, a very soft bearing metal, such, for instance, as may be made from 85 per cent. lead and 15 per cent. antimony, is excellent. This metal is, however, entirely unsuited for hard service, as it readily changes its form with increase of temperature. The bearing metal known as genuine Babbitt, consisting of tin, 85 to 89 per cent.; copper, 2 to 5 per cent., and antimony 7 to 10 per cent., is probably adapted for a wider range of use than any other metal which has ever been designed or invented. On account of the large amount of tin this metal is expensive, and there is a great temptation to palm off as a substitute a metal containing a considerable portion of lead. As a result of my experience, a considerable amount of lead can be used, provided it alloys perfectly with the other metal and does not render the compound too soft. Lead is, however, a poor conductor of heat; for a given condition of lubrication and work performed, a bearing metal containing much lead is likely to run warmer than one containing other metals.

The soft metals mentioned above possess the advantage that they can be easily melted and cast into shape in place if desired or as needed for use on the journal.

Hard Metals.

There are a number of other metals which have a high melting point and quite a large coefficient of contraction which, if used for bearing metals, must be cast in separate moulds and finished on machine tools before applying. These metals vary in hardness to a considerable extent, the phosphor bronze being probably the hardest and the yellow brasses the softest. I made extensive experiments with a bearing metal of this class consisting of an alloy of aluminum, zinc and copper, the zinc being largely in excess of

the other ingredients. That alloy was very satisfactory when zinc of the proper purity could be obtained, but was so much affected by the impurities likely to be found in zinc that it was frequently quite unsatisfactory in practice.

I have found that a mixture consisting of 50 per cent. of aluminum, 25 per cent. of zinc and 25 per cent. of tin forms an alloy which has many excellent properties as a bearing metal. It is light in weight, has a fair degree of hardness, a moderately high melting point, and, so far as I can determine from laboratory experiments and some practical applications, is a superior metal for certain kinds of bearings.

Conclusion.

From the uncertain nature of our methods of testing and from the varied conditions under which bearing metals are used, it is easy to understand the differences of opinion which are held by various engineers regarding the quality of the same bearing metal. This fact also probably explains the reason why such a variety of grades and prices of bearing metal can be marketed.

In my opinion there is no positive criterion, no single definition or specification, which can adequately describe a bearing metal which shall be universally satisfactory for all work and conditions.

Tests of Large Shaft Bearings.

Mr. Albert Kingsbury.—The following tests were made in 1904 by the Westinghouse Electric and Manufacturing Company at their East Pittsburg works. The special apparatus required was made for the Niagara Falls Hydraulic Power and Manufacturing Co., at whose request the experiments were undertaken.

The apparatus, shown in Figs. 2 and 3, consisted of a horizontal shaft supported in two bearings, each 9 inches diameter 30 inches long, with a third bearing 15 inches diameter 40 inches long, midway between the supporting bearings. The 15-inch bearing was pressed upward against the shaft by means of a lever made from two 15-inch I-beams, weighted at its outer end. The 9-inch bearings each carried half the load, less half the weight of the shaft. These three bearings are designated *A, B* and *C,* beginning at the left in Fig. 2. The shaft was driven by a Westinghouse No. 50 (150 horse-power) direct current railway motor; for shaft speeds not exceeding 500 revolutions per minute, the motor was

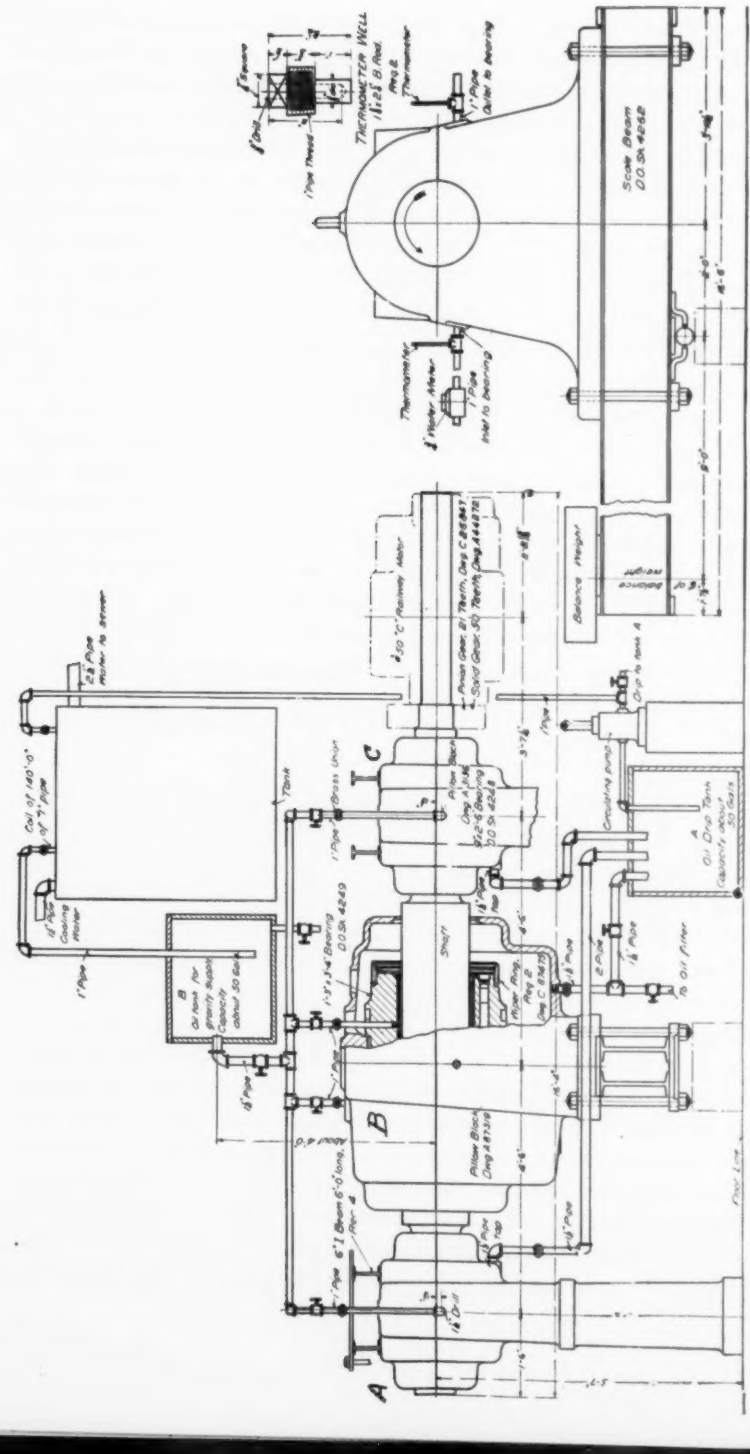

THERMOMETER WELL
Scale Beam
Balance Weight
Thermometer
Water Meter
Tank
Shaft
Pillow Block
Circulating pump
Oil Drip Tank
Floor Line

FIG. 2.

mounted as shown in the figures, with a 21-tooth and 50-tooth gear; for higher speeds the motor was mounted on the floor, with a 26-inch pulley on the armature shaft and 12-inch pulley on the test shaft, driving by an 8-inch double leather belt. The electrical power supplied to the motor was the only available basis for estimating the friction of the bearings; the motor efficiency being approximately 67 per cent. at 45 to 54 amperes, and 85 per cent. at 114 to 127 amperes, the total power consumed by the bearings *A*, *B* and *C* was determined, but their separate frictions could not be found.

The shaft journals were made true with lead laps and finished with emery cloth. The bearings were lined with genuine babbitt metal, scraped to fit the shaft. The clearance over the top and sides was about 0.03 inch in *B*, and this proved to be ample. The vertical clearance in *A* and *C* was about 0.007 inch or 0.008 inch and was not enough to provide for the expansion of the shaft and inner part of the bearings under the rapid heating of the most severe tests. The bearing sleeve of *B* was cored for water cooling; the lower half only was connected for water circulation, with thermometer wells at inlet and outlet, and with a 1-inch water meter at the inlet.

The bearings were flooded with oil from a small supply tank, to which the drip was returned by a motor-driven pump through a coil of 140 feet of 1-inch pipe, with provision for water cooling, as desired. The oil supply pipe to each bearing was connected into a wide and deep groove in the face of the babbitt, parallel to the shaft and extending nearly the full length of the bearing; the groove was located at the top of the bearing in *B*, and at the side in *A* and *C*.

Detailed data of the conditions and results of the tests are given in the appended table.

The test runs were generally of about seven hours' duration each day, starting with all parts cool, and bringing the shaft up to full speed as quickly as possible. If the speed was quickly raised to about 1,000 revolutions per minute it was found that the expansion of the shaft and inner part of the bearings caused binding, particularly in the bearing *C;* hence to reach the higher speeds it was necessary to accelerate slowly, allowing at least three hours for heating the outer parts. The load was relieved at starting because of the very great torque that would be required for starting under load.

Bearings *A* and *B* became somewhat damaged apparently from raising the speed too rapidly in the final run (No. 13), in which a speed of about 1,350 revolutions per minute was attained within

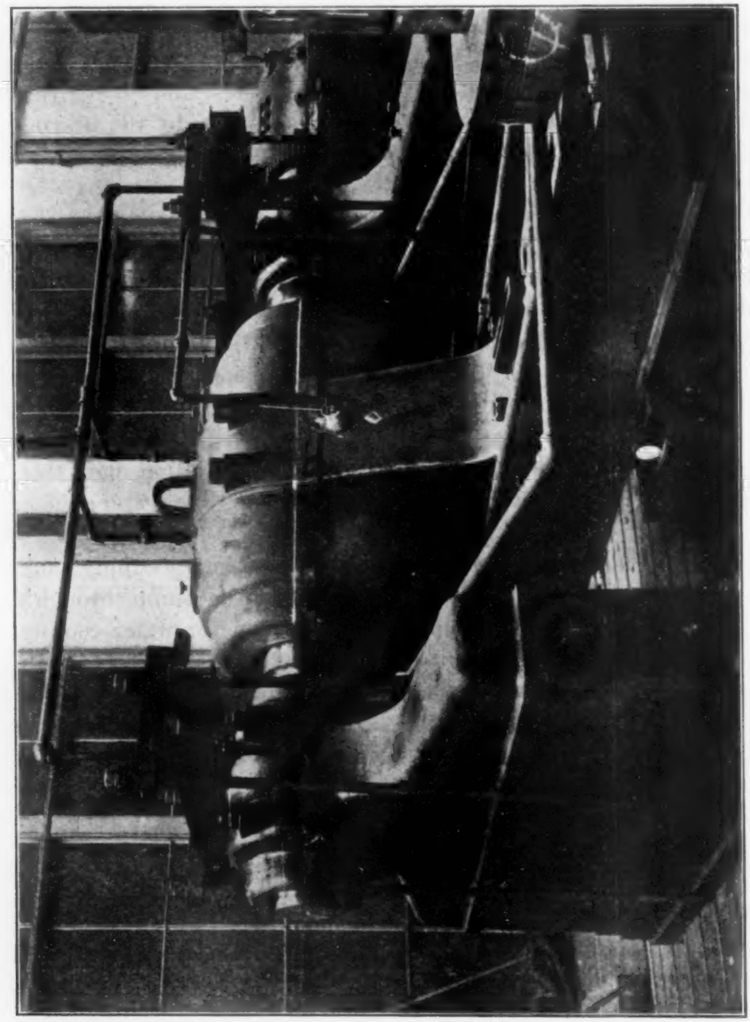


FIG. 3.

$4\frac{1}{2}$ hours, including a stop of $1\frac{1}{2}$ hours, the load being 101,000 pounds.

The thickness of the oil film between the shaft and the bearing *B* was measured approximately at successively reduced speeds

under a load of 94,000 pounds at the close of test run No. 12. Four "Bath" lathe indicators were used for this purpose, two being attached to each end of the housing, with contacts on the

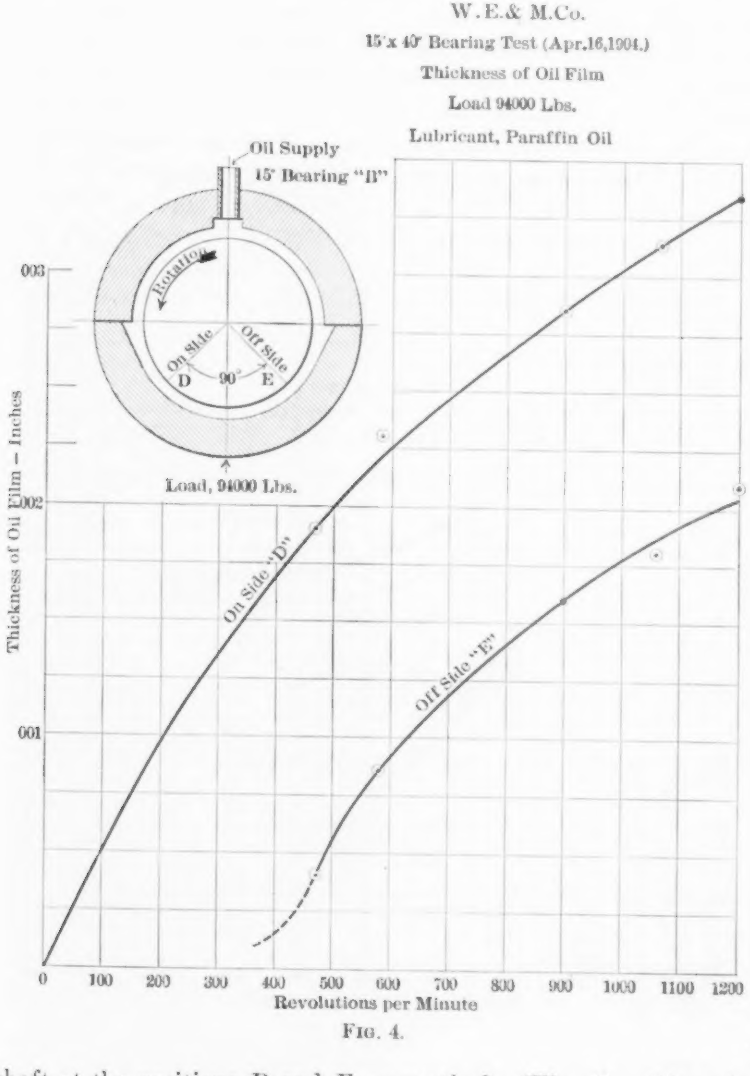


Fig. 4.

shaft at the positions D and E, respectively (Fig. 4). The relative displacement of the shaft and bearing by the oil film, as shown by the curve (Fig. 4) is greater at D than at E at all speeds; this

condition, as theoretically proved by Osborne Reynolds (" Theory of Lubrication," Phil. *Trans.*, 1886), is essential to complete lu-

W.E.& M.Co.

15"x 40" Bearing Tests

Relative Viscosities of Lubricants

By "Doolittle" Viscometer

	Spec. Grav.	Spec. Heat, (Approx.)
Heavy Machine Oil,	.92	.547
Paraffin Oil,	.89	.551

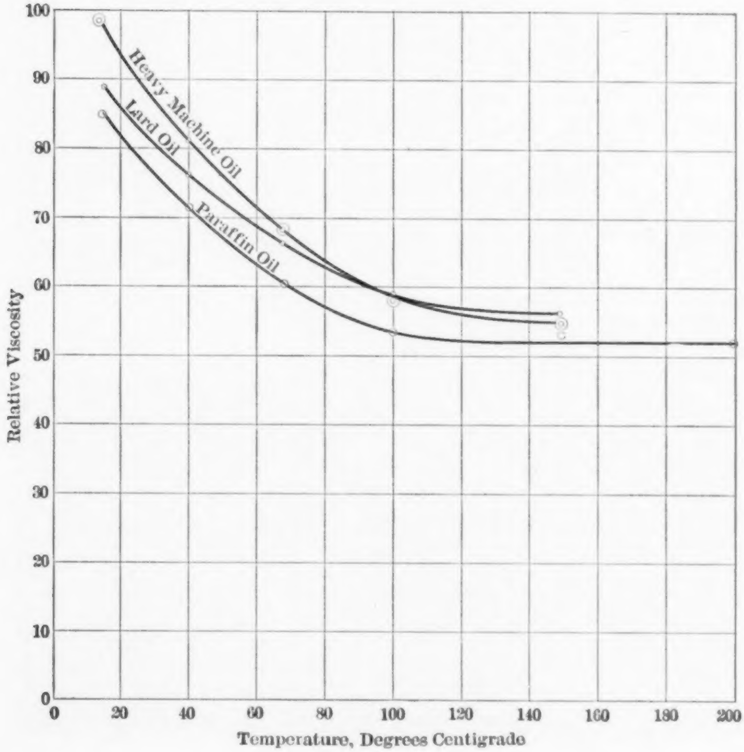


FIG. 5.

brication and is automatically maintained. It should be noted, however, that the film thickness was probably greater than the

measurements indicated, owing to the flexure of the shaft. The speed could not be controlled blow 400 revolutions per minute owing to the lightness of the load on the motor; but the rapid increase of friction due to imperfect lubrication occurred at about 300 revolutions per minute as the speed was reduced. The lower limit of speed for complete lubrication might be expected to be determined by the conditions in the bearing *A* or in *C*, or both, rather than by those in *B;* nevertheless, the curve for the point *E* appears to show that the metals in *B* came into contact at about 300 revolutions per minute, the apparent lower speed limit.

As will be seen from the table of data and results of the tests, it was found possible to run the bearings with loads and speeds

TABLE 1.—TESTS OF 15″ X 40″ AND 9″ X 30″ BEARINGS.

DATA AND RESULTS OF TESTS.

W. E. & M. CO., APRIL, 1904.

TEMPERATURES OF BEARINGS CONSTANT.*

"A" and "C" – 9″ × 30″ "B" – 15″ × 40″	Tests with heavy Machine Oil.								Tests with Paraffin Oil.		
Test Number	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**
Load on B Tons	25	25	25	25	25	25	33.6	42.3	47	47	50.5
Pressure on B lbs. per sq. in. nom. area	83	83	83	83	83	83	112	141	157	157	168
Load on each A and C Tons	11.2	11.1	11.1	11.1	11.1	11.1	15.4	19.7	22.1	22.1	23.6
Pressure on A and C lbs. per sq. inch	86	82	82	82	82	82	114	146	164	164	175
Shaft Speed — R. P. M.	300	309	506	180	179	301	454	480	946	1243	1286
Shaft Speed — Ft. per min. B	1200	1215	1990	708	704	1180	1785	1890	3720	4900	5050
Shaft Speed — Ft. per min. A and C	723	730	1190	424	422	710	1070	1030	2220	2930	3039
Motor Amperes	53	54	57	51	45.5	49.5	51	53	122	114	117
Volts	258	260	428	141	126	227.5	350	379	301	368	392
Electrical H. P.	12	18.9	32.7	9.68	7.7	15.1	24	26.9	49.3	56.3	61.6
Friction H.P. Total for A B & C	16	12.6	21.7	6.43	5.12	10.1	16.	17.9	41.9	47.8	52.3
Friction Torque lbs. ft. Total A B & C	201	214	225	188	150	176	185	196	233	202	213
Average coeff. of Friction for A B & C — Starting (cold)	.12	.146	....	....	....	....	....	....	....	....	....
Average coeff. of Friction for A B & C — Running	.0044	.0045	.0048	.0040	.0032	.0037	.0029	.0024	.0025	.0022	.0022
Temperat. Centigrade — Atmosphere	17	18.7	20	20.4	20	16.8	17.3	19.7	20.5	20	19
Temperat. Centigrade — Top Sleeve of Bearing A	48.8	44.9	54	35.8	42	44	52.2	52.9	58.7	74	97
Temperat. Centigrade — Top Sleeve of Bearing B	....	....	....	....	....	....	....	....	....	67	58
Temperat. Centigrade — Top Sleeve of Bearing C	47.5	45.9	55	40.5	45	45.1	57.8	54.9	67.2	91	117
Temperat. Centigrade — Oil Supply	24.6	12.4	17.7	9.4	32.5†	11.7	14.7	16.4	21.7	20.6	16.5
Temperat. Centigrade — Oil Drip A	....	....	....	....	....	....	46.9	47.	58	72	65
Temperat. Centigrade — Oil Drip B	....	....	....	....	....	40.7	49.2	51.3	56.	61.5	54.
Temperat. Centigrade — Oil Drip C	....	....	....	....	....	....	46.8	49.5	59.5	66	62.5
Temperat. Centigrade — Cooling Water for Bearing B — Supply	7.	6.3	7.6	8.1	19.2	10.9	10.9	11.8	11.	12.	16.
Temperat. Centigrade — Cooling Water for Bearing B — Discharge	18.	13.4	14.6	16.3	32.6	37.9	47.2	48.8	38.5	69.6	69.
Temperat. Centigrade — Cooling Water for Bearing B — Rise of Temp.	11.	7.1	7.	8.2	13.4	27.	36.3	37.	27.5	57.6	53.
Lbs. Oil Supplied per min. A	....	4.	6.2	3.1	3.9	4.4	6.4	8.2	1.2	9.4	2.8
Lbs. Oil Supplied per min. B	....	11.6	14.1	3.1	10.9	4.0	3.1	5.8	10	7.6	5.9
Lbs. Oil Supplied per min. C	....	2.3	3.5	1.5	1.8	2.7	3.8	5.	8.5	9.9	10.
Lbs. Cooling Water per min to B	9.6	13.7	24.5	9.1	.3	.55	.92	.49	6.2	.3	.3
Per cent. of total heat carried from A B & C — By Oil	....	....	....	....	....	....	63.	80	61.2	61.	36.
Per cent. of total heat carried from A B & C — By Water	22.3	19.6	20	29.5	2.	3.7	5.3	2.6	10.3	.92	.76
Per cent. of total heat carried from A B & C — By Air and Radiation (by difference)	....	....	....	....	....	....	31.7	17.4	28.5	38.1	63.2

* In 13th conditions were not constant; the data given are those taken just before the bearing seized at speed 1350 R.P.M.

† Oil supply was not water cooled.

greatly exceeding the ordinary values in practice, even without water-cooling the bearing sleeve. The extreme case of a successful test is shown in column 12, in which a run was made under 94,000 pounds' load at 1,243 revolutions per minute with only a very small amount of water run through the bearing sleeve, merely for determining its temperature. The oil supply, however, was cooled in all the tests except No. 7 and the first half of No. 13.

It may be noted in the table that the power required to drive the shaft was very nearly proportional to the speed of rotation, independently of all other conditions, such as load, final temperature attained and lubricant used. This result is probably not a general law, but is noteworthy even as an accidental relation.

Water as a Lubricant for Machinery Bearings.

Mr. C. W. Naylor.—Two jack shafts, turned steel, 5 inches in diameter and 18 feet long, each running at 250 revolutions per minute, receiving and transmitting each 175 indicated horse-power from a pair of steam engines, and driving by leather belts 5 electric generators.

Bearing boxes 14 inches long, plain, non-self-oiling type and

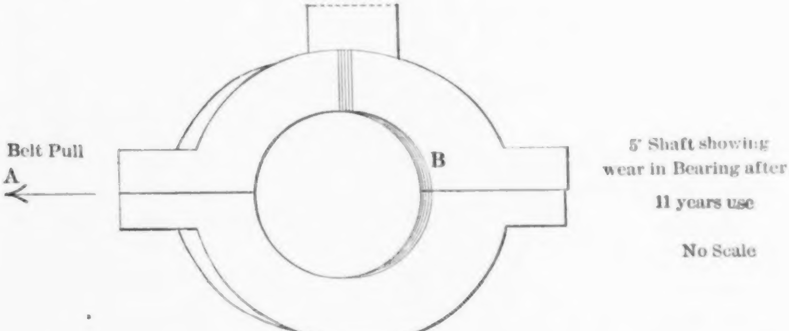


Fig. 6.

split on a horizontal plane. The pull of the six belts were all in the same direction and horizontal, or practically so, the shaft necessarily pulling directly against the split in the boxes, which were cast iron babbitt filled.

.For two years 1888-89, much trouble was experienced in keeping the boxes cool with oil, of which several different grades were

used, as well as greases. In 1890 water was tried on the bearings—Lake Michigan hydrant water—with such good results that it was continued until 1901, for a period of eleven years, without any serious interruption for ten hours each week day. A small stream of water was allowed to trickle through the bearings, and at five minutes before closing down oil was fed into same to prevent the shaft rusting and sticking over night when the machinery was not in use. The wear on the babbitt and boxes for the eleven years was about one-fourth of an inch and on the shaft it was nil. On account of the shaft being pulled to one side all the time, the

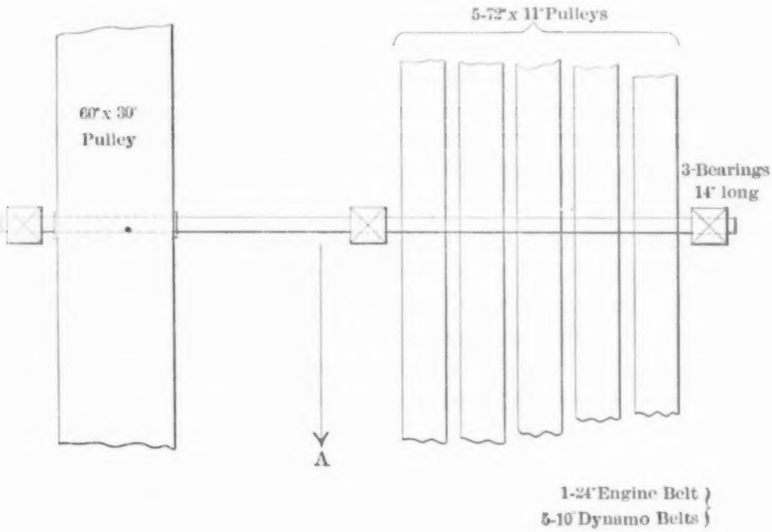


FIG. 7.

wear left a space back of the shaft in the boxes in which an ordinary pocket rule could be inserted.

The temperature of the engine-room was about 100 degrees Fahr., and of the water from 45 degrees in winter to 65 degrees or more in summer. The increase in temperature of the shafting was inappreciable, and less trouble on that score was experienced with eleven years' use of water than with one month's use of oil.

The strain or pull on the shaft was not measured, but was very great, for the reason that the belts on the dynamos were oil-soaked nearly all the time and had to be run very tight.

The plant was abandoned in 1901.

Marine Thrust Bearings.

Mr. G. W. Dickie.—I was somewhat surprised when asked to give a short paper as food for discussion before the Society on the above subject, for, judging from the rapid introduction of the steam turbine for ship propulsion, the thrust bearing which has played so important a part in marine steam engineering is in danger of losing its importance in marine engine design, the thrust of the propeller on the one end of the shaft being balanced by the thrust of the turbine on the other end. This defect in the Parsons turbine, the end thrust, is a great advantage when applied to screw propelled ships, for it balances the thrust of the screw, thus saving the very material loss by friction of the thrust bearing and renders the Parsons turbine an ideal form of motor for ships. Still for some time to come we will require thrust bearings for marine engines, and an inquiry as to what the main features in a well-designed bearing of this character should be time well spent.

The marine thrust bearing has not changed in type since the introduction of screw propulsion, although many attempts have been made to introduce some kind of ball or roller bearing with the object of reducing the friction of the extended rubbing surfaces of the bearing, but no one of such devices has succeeded in permanently taking the place of the collar and ring thrust. The adoption of the horseshoe ring, better workmanship and improved methods of lubrication, together with water cooling devices, have brought the marine thrust bearing to its present efficiency.

The function of the thrust bearing renders it difficult to experiment with for data, it has therefore developed along the line of meeting pressure with surface, providing that all the surface involved will take load, lubricating and watching the result. In computing the pressure on a thrust bearing, the usual practice is to find the indicated thrust, assuming two-thirds of the indicated horse-power to be effective, the result will be somewhat in excess of the actual thrust even in the best cases, the pressure on thrust bearings being

$$P = \text{I. H. P.} \times \frac{2 \times 60 \times 33{,}000}{S \times 3 \times 6080}.$$ P = Pressure on thrust bearing S = Speed of ship in knots per hour. Applying this to

several types which cover general practice and which I have taken from my own work which proved to be satisfactory.

No. 1. Armored cruiser. Speed	22 knots.
Thrust ring surface, horseshoe type	1188 square inches.
Horse-power, one engine	11500.
Indicated pressure or thrust on bearing	112,700 pounds.
Pressure per square inch of surface	95 pounds.
Mean speed of bearing surfaces	642 feet per minute.

This would fairly represent naval marine engine practice, and under the full power conditions as given above the thrust bearing needs careful watching. The length of the bearing proper, that is, from the after face of the after horseshoe to the after face of the forward horseshoe being 72 inches, the danger is that any part of the bearing getting hot may so change the relative distance between the shaft collars and the thrust rings as to disturb the distribution of load and leave part of the bearing out of service. I think that I was the first to make the horseshoe rings with an independent circulation of cooling water through each ring.

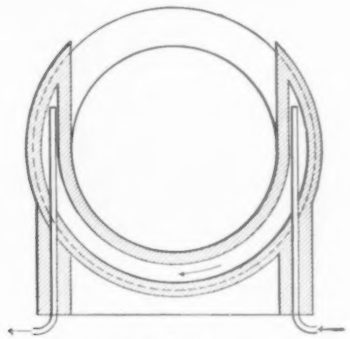


FIG. 8.

This enables the attendant to control the heat of each ring independently and thus maintain an even temperature throughout the whole bearing. The base of the bearing forms an oil reservoir, the bearing at each end having a stuffing box so that the oil can be carried above the lower line of the shaft, a cold water coil is fitted in the oil chamber to keep down the temperature. I find that a small surface is sufficient for this purpose where the horeshoe rings are fitted with water circulation. The bearing noted above has 15 feet of cooling surface for the lubricating oil.

No. 2. Protected cruiser. Speed	22½ knots.
Thrust ring surface, horseshoe type	891 square inches.
Horse-power of one engine	6800.
Indicated pressure or thrust on bearing	89,000 pounds.
Pressure per square inch of surface	100 pounds.
Mean speed of bearing surfaces	610 feet per minute.

This bearing was fitted with independent water circulation through each horseshoe ring like No. 1, and with 10 feet of cooling surface for the lubricating oil, the working proving very satisfactory, the thrust bearing requiring no more care than the other bearings of the engine.

No. 3. Torpedo-boat destroyer. Speed	28 knots.
Thrust ring surface, complete rings	581 square inches.
Horse-power of one engine	4200.
Indicated pressure or thrust on bearing	33,600 pounds.
Pressure per square inch of surface	58 pounds.
Mean speed of bearing surfaces	827 feet per minute

This bearing was fitted with a complete water jacket and required a large stream of water to control the temperature at full power, and constant care was necessary during a two hours' full power run. It will be noted that the pressure per square inch of surface was much less than that of No. 1 or No. 2, but the surface velocity was much higher and therefore lubrication was more difficult.

No. 4. First-class modern passenger ship. Speed	21 knots.
Thrust ring surface, horseshoe rings	2268 square inches.
Horse-power of one engine	15000.
Indicated pressure or thrust on bearing	154,500 pounds.
Pressure per square inch of surface	68.1 pounds.
Mean speed of bearing surfaces	504 feet per minute.

In this case there is no independent water circulation through the horseshoes, and the bearing gives no unusual trouble. It will be noted that in this case the bearing surface speed is less than in the others, and consequently lubrication is more certain.

There is considerable difference in the practice of prominent marine engine builders in regard to thrust bearings, but generally it will be found that for steady work the pressure per square inch of surface should not exceed 75 pounds with a speed of 500 feet per minute.

Mr. Joseph J. White.—The journal box hereinafter described

was designed to overcome the difficulty of adjusting the cap of the box in general use.

Twenty years ago I was interested in the manufacture of wood-working machinery, the journal boxes of which were usually babbitted for shafts making from four thousand to five thousand revolutions per minute. Paper or wood liners were placed beside the shaft in the box to support the cap, which was held in place by four bolts. To stop each of these bolts at the proper point was no easy matter, and there was no way of knowing whether the feat had been successfully accomplished without starting up the machine and noting the result, then twisting and trying again and again until the shaft ran without heat or vibration.

To overcome these difficulties I constructed and patented during 1891 the journal box shown in Fig. 9. Briefly stated, the im-

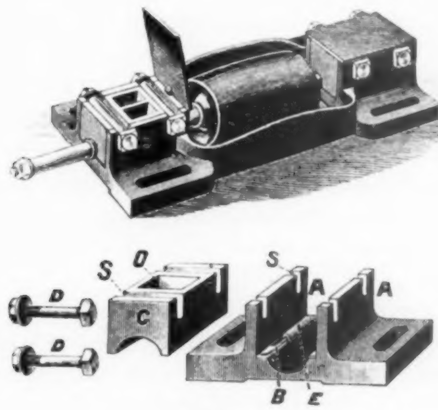


FIG. 9.

provement consists in fitting the cap "C" accurately between the side walls "AA," where it may be quickly and securely clamped by the bolts "DD."

Perfect adjustment is obtained by first oiling the shaft and then placing the cap "C" in position, where it rests upon a film of oil. The side walls and cap are then clamped together without changing the position of the cap, thus leaving it in adjustment with the proper oil space at all points and no possibility of heating if oil is supplied.

During the last twelve years these boxes have been used successfully on wood-working machinery.

Mr. George M. Basford.—The unknown quantities in the matter of stresses to which locomotive parts are subjected place locomotive designing in a class by itself.

The rules for designing certain parts are absolutely fixed and definite, in accordance with well-known principles, while the designing of other parts depends upon quantities which are uncertain and impossible to measure.

In the matter of locomotive bearings, it is of little practical use to study the coefficient of friction. Rules for bearing pressures which are entirely satisfactory for other construction will not answer at all for locomotives.

Bearing areas for locomotive journals are determined chiefly by the possibilities of lubrication. They are affected by the very severe service to which locomotives are subjected, and the presence of dust, sand, ashes and cinders must be reckoned with.

In stationary or marine practice definite loads may be provided for, but this is not the case with locomotives. The fluctuating load, combined with the possible presence of grit, renders it necessary to provide for sudden changes in the coefficient of friction by providing a large factor of safety.

Locomotives must work out of doors with varying temperatures, and they must work under conditions which render it impossible to give them the attention which all kinds of expensive machinery should receive.

Concerning locomotive bearings, an extended experience has shown that crank pins may be loaded to from 1500 to 1700 pounds per square inch. These bearings are subject to alternating stresses, rendering lubrication relatively easy, and lubrication is the limiting factor in locomotive bearings.

Wrist pins may be loaded to about 4000 pounds per square inch because their rotary motion is not complete and the thrust changes twice in every revolution. With journals the case is different, there being no relief to permit of easy lubrication. On locomotive driving journals it has been found that the following figures give good service:

Passenger locomotives,	about	190	lbs.	per	sq.	in.
Freight	"	"	200	"	"	"
Switching	"	"	220	"	"	"

To provide the proper bearing area, and also provide for the piston thrust, large modern locomotives sometimes require main driving journals as large as $10\frac{1}{2} \times 13$ inches.

Car and tender journals present the condition of beams fixed at one end and loaded more or less uniformly. In these cases two limitations to the sizes of bearings are presented:

First, the fiber stresses of the journal must not be too high.

Second, there must be sufficient bearing area to insure cool running.

As a rule the various sizes of axles, adopted as standard by the Master Carbuilders' Association, may be loaded slightly more in pounds per square inch of projected bearing area without exceeding the allowable fiber stresses, than would be permissible to provide properly against heating, but both of these limitations must be borne in mind.

Car and tender bearings are usually loaded from 300 to 325 pounds per square inch of projected area, but even this unit load is misleading, because in parts the load per square inch of actual bearing area may be very much higher because of the rough character of the bearing.

Mr. P. H. Been.—The following represents the practice of one of the largest Corliss engine builders in the United States as used for engines which are either belted for power purposes or direct connected to generators:

1. Metals suitable for bearings:

All bearings, no matter of what pressure or speed, are lined with the best grade of babbitt metal.

This babbitt metal covers the entire bore of the shells.

The babbitt is held by dove-tailed grooves, the grooves being cored in the cast-iron shell

After babbitt is cast in shell, it is hammered in place with a peen hammer and then bored inch larger in diameter than the shaft.

2. Lubrication of bearings:

The oil box on bearings is fast going out of use. On most engines two to four (depending on length of bearing) pressure oil cups are used.

The Lonergan or Lukenheimer pressure oil cups are most generally used.

On a few engines oil has been fed under the shaft by pressure; but this, however, is not the customary way of oiling bearings.

The standard method is to place the oil cups on the cap of the

bearing, and the oil drops from the cup into a cored pocket, in the top shell, this pocket being filled with waste.

Most engineers flood their bearings with a mineral oil, using the same over again by a gravity oiling system.

The amount of oil used depends on conditions. A 22 and 42 x 42 tandem compound engine, directly connected to a generator, running 100 revolutions per minute with 120 pounds steam pressure, had 50 gallons of oil pass over the main and out board bearings a day.

This engine had a variable load and ran 20 hours daily, with bearings having a temperature of 124 degrees, engine room being 92 degrees Fahrenheit.

3. Method of cooling bearings:

Bearings of standard construction have no means of being cooled.

In special cases the lower shell is cored so that water can be run through it.

In most stations where the lower shells have been cored for water jacket it has been found unnecessary to use it.

4. Limits of speeds and pressures:

The greatest pressure per square inch of projected area allowed for all shafts is 140 pounds.

The speed of these shafts runs between 75 and 150 revolutions per minute.

The weights considered on the bearing are the dead weights only; that is, the wheel, generator rotor, shaft, cranks and eccentrics.

Steam pressure is not considered.

On most engines we find that the pressure per square inch, multiplied by the circumference of shaft in feet, multiplied by the revolutions per second, lies between 1,000 and 1,300.

5. Designs of bearings for high speeds and high pressures:

Bearings for horizontal engines are generally made in four parts, 1 bottom, 2 sides and 1 top shell.

The cap is independent of the shells.

The side shell, on the side away from the cylinder, is adjustable by two set screws.

The other side shell is adjustable by shims.

The bottom shell bears on a center rib only, which allows it to rock with deflection of shaft.

Bearings for vertical engines are in two parts, one lower and one upper shell.

The lower shell is a ball and socket, which allows shell to rock with deflection of shaft.

The upper shell is cast with the cap.

There is no way provided for taking up the wear of these shells.

It has been found unnecessary as all the wear is down.

When shell has worn down a great amount, shims are taken out of the base of the generator, which allows the rotor to come central with the yoke.

6. Thickness of oil film or allowance between journal and bearing:

There is no information about this for Corliss engine bearings to my knowledge.

7. Thrust bearings:

Not used for Corliss engines.

8. Ball bearings:

Not used for Corliss engines.

9. Roller bearings:

Not used for Corliss engines.

Mr. Henry Hess.—My contribution to the general discussion of bearings will be confined to a necessarily rather brief and cursory review of ball and roller bearings, and a slightly more detailed reference to ball bearings of a type with whose development and operation I am personally familiar. As the rules limit each individual contribution to a thousand words or so and to only ten minutes of time, most of this meager allowance will be devoted to the presentation of examples from actual practice.

An adequate presentation satisfactory to the analytical engineer is naturally impossible under the rules.

Both ball and roller bearings are of hoary origin. The rollers used under a block of stone by the ancient Egyptians and in exactly the same way under the skids of a crate containing the latest product of the modern machine shop show familiarly the advantage of the substitution of rolling for sliding friction.

The early recognition of this advantage led naturally to many attempts at the employment of rolling elements in the journals of machines. Until relatively recent days such attempts have been chiefly failures—interesting—but failures nevertheless. The causes were simply imperfections in the shape of the rolling elements and their supporting surfaces, resulting in the loads being actually imposed on insufficient areas, though, theoretically, greater ones were provided. Then, too, the laws for securing true rolling and eliminating all sliding have been formulated but relatively recently, so that even at this day it is not uncommon to find thrust bearings with cylindrical rollers, that necessarily can have pure rolling at only a single girdle in their length, while all other zones must slide, since only balls or cones whose apices lie in the center of revolution can roll truly.

The bicycle is responsible for the widespread realization of the possibilities of the saving of work by ball bearings; without these it probably would never have had the vogue it acquired; the rider who remembers the difference in freshness at the end of a half day's tour on a wheel fitted with cone bearings and one fitted with ball bearings can bear eloquent witness. Simple as is the design of a ball bearing of correct action, yet but few gave proof of anything but mistaken ingenuity. The saving element in the bicycle was the light load. The hopes cherished and translated into practice for more ambitious mechanisms soon resulted in failure. This was attributed to insufficient bearing surface. Further development was along two lines: One body of constructors fell back on rollers, with the idea that a roller had line and therefore more support than a ball with its theoretical mere point; others multiplied the number of balls.

Some years ago Professor Stribeck, the able head of the Technical Laboratories at Neu-Babelsberg, near Berlin, undertook for one of Germany's leading industrial establishments a thorough investigation of the entire subject of the relation of sliding, roller and ball bearings. The maximum loads under which the temperature remained constant was determined. There was also considered the time that a bearing had to be run under a given load until such constant temperature was attained.

Part of the results are given in Fig. 10 in the relation of the coefficient of friction (referred to the shaft diameter) to the specific loads. For any sliding bearing the specific load is the pressure in kilograms per square centimeter of projected journal area; for

roller bearings one-fifth of the number of rollers times the length and diameter of the rollers is considered as the equivalent of the

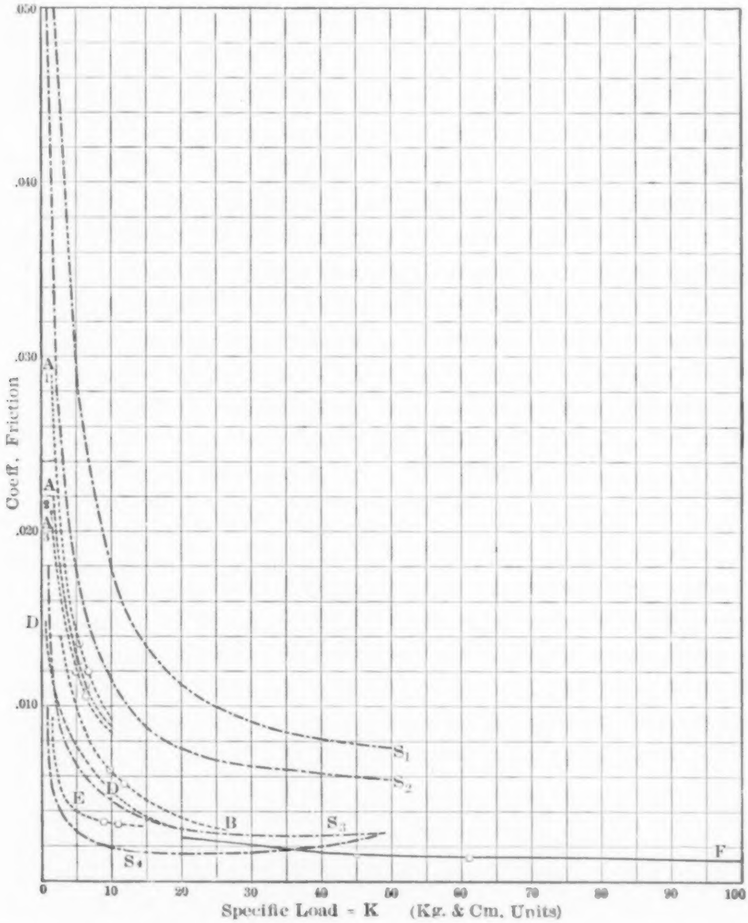


FIG. 10.—PLAIN, ROLLER AND BALL BEARINGS. RELATION OF THE COEFFICIENT OF FRICTION (REFERRED TO THE SHAFT DIAMETER) TO THE SPECIFIC LOAD. TESTS MADE BY PROFESSOR STRIBECK.

The specific load is the load per unit of the carrying element; for plain bearings such element is the projected journal area or: $K = LD$.

For roller bearings the projected area of the carrying rollers (one-fifth of the rollers only is considered as carrying) is $K = \frac{ZLD}{5}$. For ball bearings the squared ball diameter of all carrying balls is taken (one fifth of the balls only are taken as really carrying) $K = \frac{ZD^2}{5}$.

Curves "S" are for a babbitted journal—S_1 @ 1100 R.P.M., S_2 @ 380, S_3 @ 64 S_4 @ 12. Curves "A" are for rollers in a bronze cage circumferentially fitting and supporting each roller throughout its length. A sleeve on the shaft and a liner in the box carry the rolls. Hard and soft rollers were tested. A_1 @ 1100 R.P.M., A_2 @ 285, A_3 @ 190. Curves "B" are for hollow rollers on loosely fitted pins that connect the end cages. No liners are used on the shaft or in the box. R.P.M. from 190 to 760. Curves "C" are for hollow flexible short sheet metal rollers on loosely fitted pins that connect the end cages. No liners used on the shaft or in the box. 1100 R.P.M. Curves "E" are for flexible spiral rollers; the rollers are loose and are aligned by three longitudinal bars connecting the end cages. No liners for shaft or box are employed. 56 to 1100 R.P.M. Curves "F" are for a ball bearing of the two point curved ball race type at 65 to 1150 R.P.M.

The range of permissible specific loads, as indicated by the appearance of the bearing after long continued runs, is shown by small circles on the curves.

projected journal area; for ball bearings one-fifth of the number of balls times the square of the ball diameter is considered as such equivalent. Within the limits of numbers of balls and rollers in an ordinary journal one-fifth of the number may be considered as practically sustaining the load, though naturally not equally. For roller and ball bearings angular velocity has not much influence within quite wide limits; wider for balls than for rollers; note the revolutions per minute figures of the curves.

Noticeable is the very low coefficient of friction of the roller —and more especially of the ball-bearing. A further marked characteristic that these tests developed was the value of the coefficient of friction at rest; with roller bearings the latter was far less than with plain bearings, and for the ball bearings the friction of rest and motion was found to be identically low; this is a very important quality in all machines that must be frequently started and stopped, as in suburban and urban train and car service.

Stribeck concludes that for continuous service with roller bearings K may lie between 6 and 11 Kg.

For lineshafts employing soft rollers the journal length works out but slightly less than the usual plain box length; K should lie well toward the lower limit. For hardened rollers on hard surfaces K may lie toward the higher limit. It is important that the rollers should be parallel to the shaft when under load. Tests of various cage constructions and of free rollers have shown that this is attained most nearly when the rollers are not tied together but are left free to adjust themselves; lack of truth will cause a slewing of the rollers as they pass under the load; as they pass out to the free side of the journal they can again straighten out unless confined in rigid relationship to those still under load.

At first glance the relatively high carrying capacity and low friction of the ball bearing as compared with the roller seem strange. The explanation is not far to seek. For pure rolling there would be no difference between ball and roller; but pure rolling is a theoretical possibility merely, requiring for its realization absolutely true shapes initially, and inelastic materials that will not change shape under load. To produce a series of rollers that are truly cylindrical alike as to diameters and also a cylindrical shaft for them to roll on, and a truly cylindrical box to roll in within the requisite small limits of error, is difficult and commercially not realizable with existing shop facilities. More serious still is the fact that under heavy loads even such accuracy as may be had is largely defeated by the deflections of the machine fram-

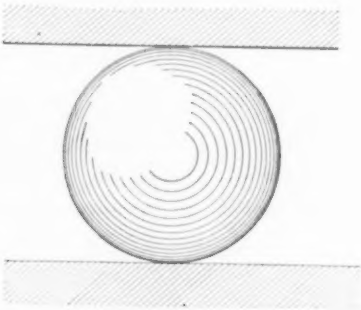


FIG. 11.

ing. The familiar tendency of rollers to skew is the result of such lack of truth or deflections. It follows that the theoretical line contact of the roller does not exist in fact, but is limited to but a short portion of the roll length.

Difficult as the production of a true ball is under ordinary shop conditions, specialists in the manufacture of this article can deliver balls at relatively small cost that are true spheres to within less than one-ten-thousandth part of an inch. This fact suggested the substitution of a large number of balls for a small series of rollers in the solution arrived at by Simonds of Fitchburg; there remained still the difficulty of the lack of truth of the shaft and box; to make up for this by selecting balls to suit successive zones was a way out, but hardly a commercially practicable one. Many constructions were devised for rocking and compensating mountings that would also take care of the difficulties introduced by

deflection under load of the supporting shaft and box. Finally a German engineer, Mr. Riebe, decided that the proper thing was to employ but a single row of balls and so have a journal of no appreciable length that therefore could not cause trouble by de-

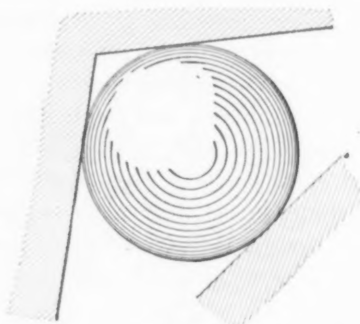


FIG. 12.

flection, and then to proportion the diameter and number of balls to the load to be carried. As the number of balls in a single circle was necessarily limited, and as the journal diameter could not be indefinitely increased, it became important to develop

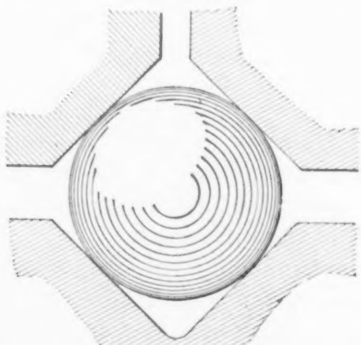


FIG. 13.

the carrying capacities as affected by the shape of the ball tracks and the nature of the materials. Tests were made of ball bearings of various shapes and relationship of contact. It was found that the frictional resistance was least for balls rolling between straight line sections, Fig. 11, of the two-point order of contact. It was

further found that increasing the points of contact to three and four, Figs. 12 and 13, gave higher frictional resistance, due no doubt to the slight variation from the correct relationships of the con-

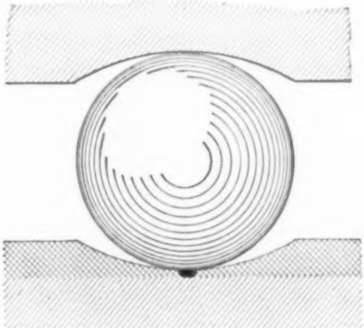


FIG. 14.

tact points with the centers of revolution of balls and journal; the carrying capacity was not materially different for these forms. Theoretically, No. 13 should carry rather more, but the greater

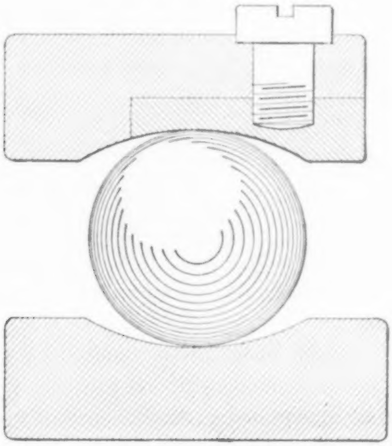


FIG. 15.

tendency to partial sliding, as compared with the pure rolling of Fig. 11, partially offsets this.

Curving the race, Fig. 14, resulted in materially greater carrying capacity with but barely measurable increase in friction. This greater carrying capacity is accounted for by the fact that

in a rolling bearing, in the absence of grit, there is no wear as in a sliding bearing, but a destruction of the surfaces by crushing, indicated by the heavy dot. The point in the race that is overloaded is flaked out. If, in a ball bearing, the race section is curved, the stressed particle in Fig. 14 cannot get out, being supported and confined by the wedges of material on either side. These wedges are indicated by cross hatching.

Naturally, as the carrying capacity of a bearing with curved race sections was greatest, it followed that any openings for side filling of the balls which would break such section, and therefore reduce the capacity, must be kept at the unloaded side or avoided altogether. Riebe's design, Fig. 15, was along the former lines; the ball was introduced through a cut in one race at the side; the continuity of the race was restored by a clip; but still the joint was a weakness, so that the screwhead was utilized to keep this joint at the unloaded side. Later a Mr. Conrad suggested that it would be better to have no weak point, but that both races be of uniform section throughout; better still, he also accomplished this by filling in as many balls as would fill one-half of the bearing, assembling this, distributing the balls and then inserting separators that could maintain such distribution and so lock the whole into a complete unit. The lesser number of balls was compensated for by larger sizes and by the possibility of somewhat higher loading under most conditions owing to the absence of certain interferences incidental to every ball-bearing having balls in unyielding contact.

Further experiments—again carried out by Professor Stribeck—on bearings and balls of various sizes, and races of various degrees of curvature and many standard and special steel alloys, developed data for the reliable designs of bearings suited to all sorts of conditions.

That these two constructors have, backed by the researches of Professor Stribeck, been successful to the point where the most conservative need no longer shy at the employment of ball bearings under any conditions, is best proven by the consideration of a few typical examples from actual practice. As the mention of firms is more or less out of place in a discussion such as this, no names appear, but it may be said that the various prints shown all represent constructions that have been actually carried out and are standard practice with firms of international reputation.

Fig. 16 is a continuous annular race in perspective; it consists of an inner and an outer race both provided with curved ball tracks that are uniform and unbroken around the complete circumference and of a series of balls separated by elastic distance pieces, which latter are usually provided with felt plugs for storing up lubricant.

The first example shown is the mounting of a tubular cement

Fig. 16.

mill, Fig. 17. This is also interesting as a journal of somewhat unusual dimensions—39 inches—and will carry safely 40,000 pounds of weight in such a machine.

In a pillar crane, Fig. 18, of a certain French type the whole load of 39,000 pounds is hung on top of the post and sustained by a ball thrust making use of $1\frac{3}{4}$-inch balls. Immediately below it a radial bearing takes the 21,000 pounds of horizontal thrust on $1\frac{5}{8}$-inch balls.

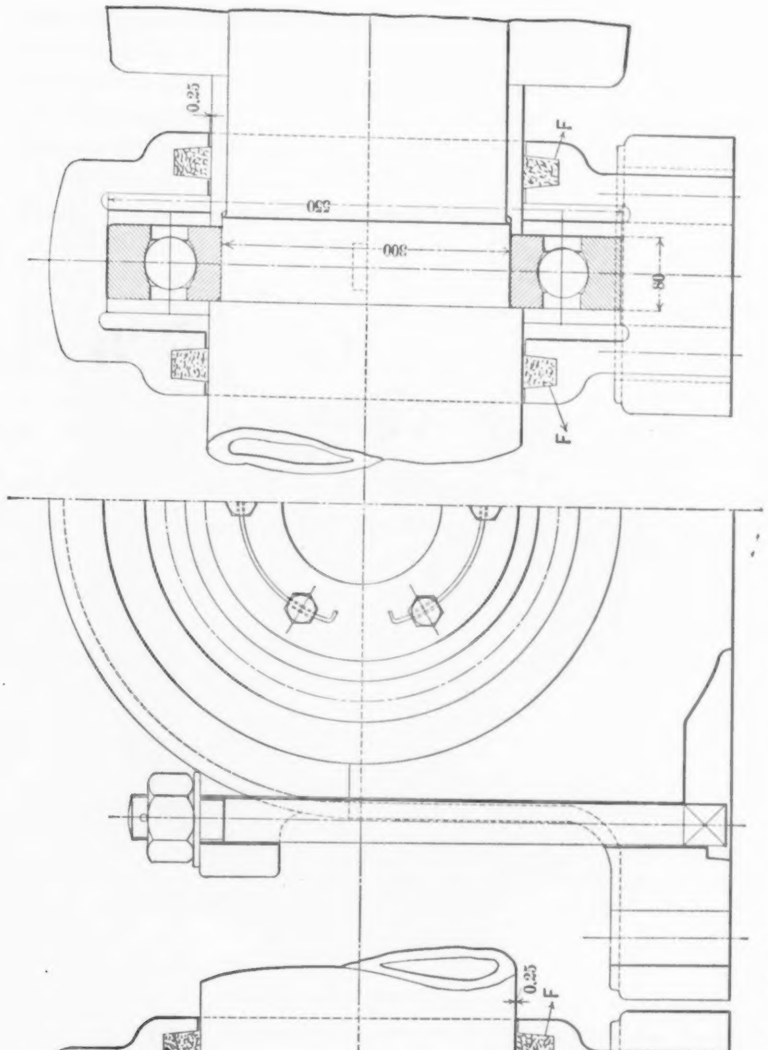


FIG. 17.—MILLING MACHINERY FOR CEMENT. 40,000 LBS. BEARING LOAD.

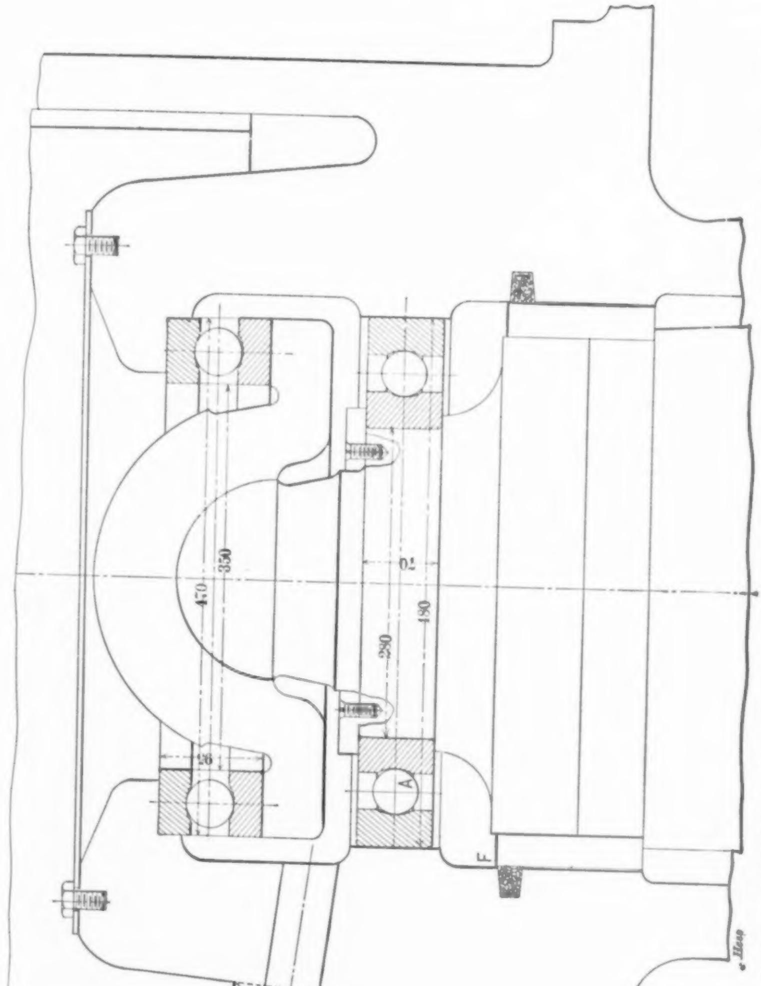


FIG. 18.—PILLAR CRANE. 39,000 LBS. VERTICAL AND 21,000 LBS. HORIZONTAL THRUST ON THE BALL BEARINGS.

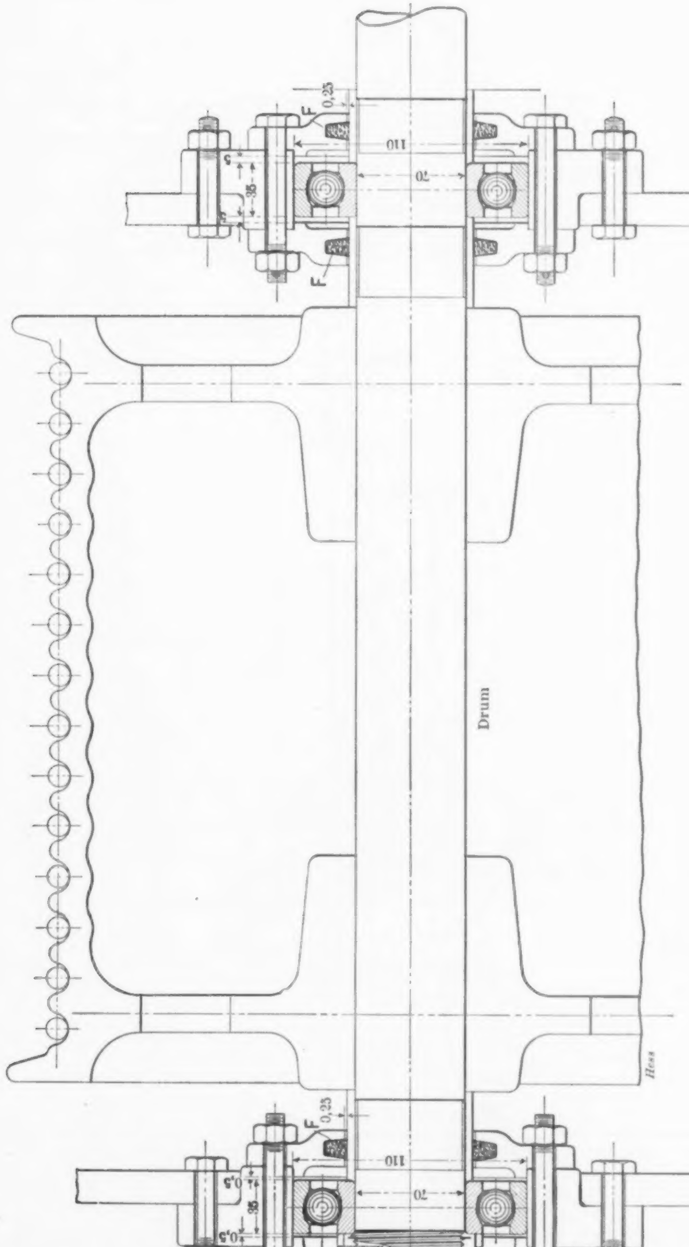


FIG. 19.—HALF-TON CRANE DRUM. 285 FT. P. MIN. LIFT.

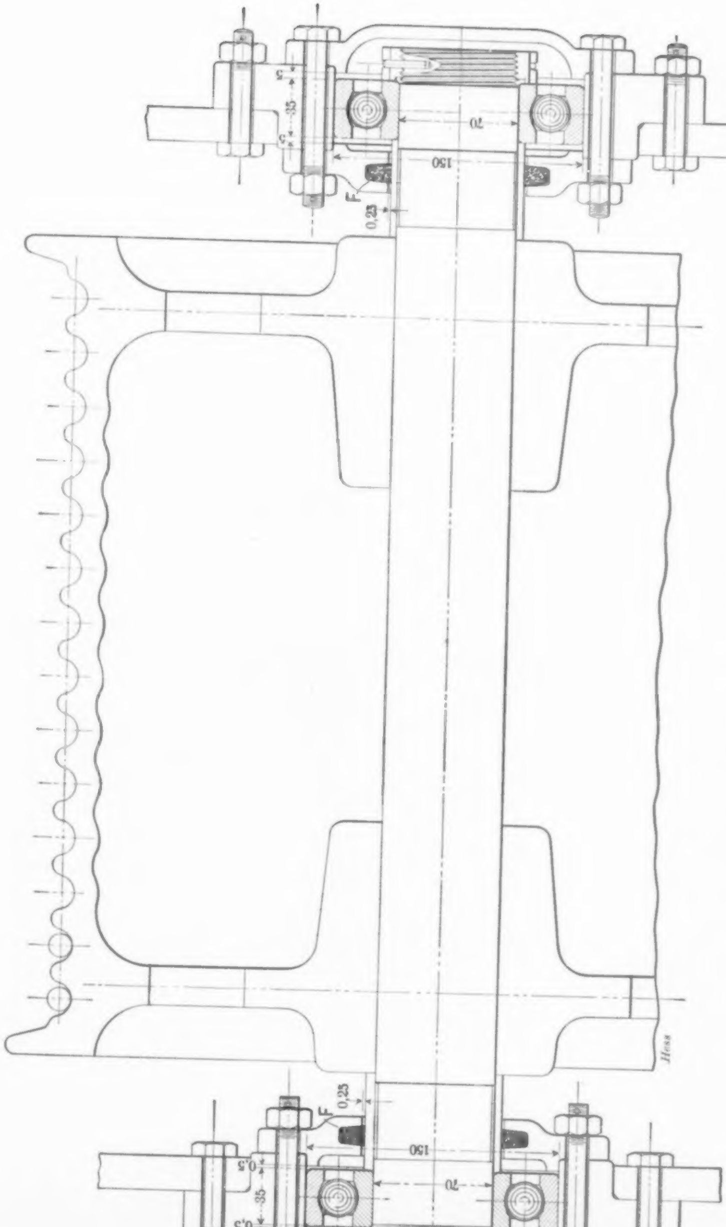


FIG. 20.—2½-TON CRANE DRUM. 185 FT. P. MIN. LIFT.

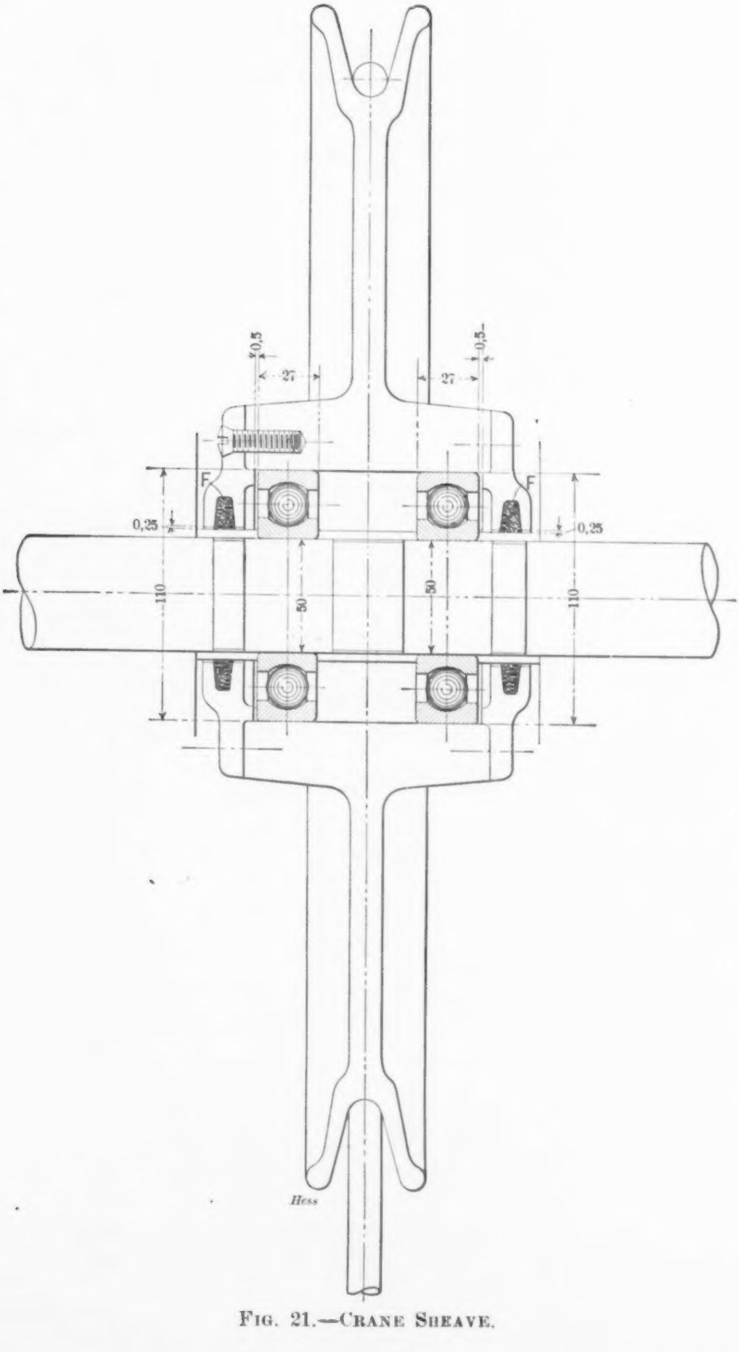


FIG. 21.—CRANE SHEAVE.

A traveling crane may suitably follow; the light and heavy drums and sheaves, as well as the crane hook, Figs. 19, 20, 21

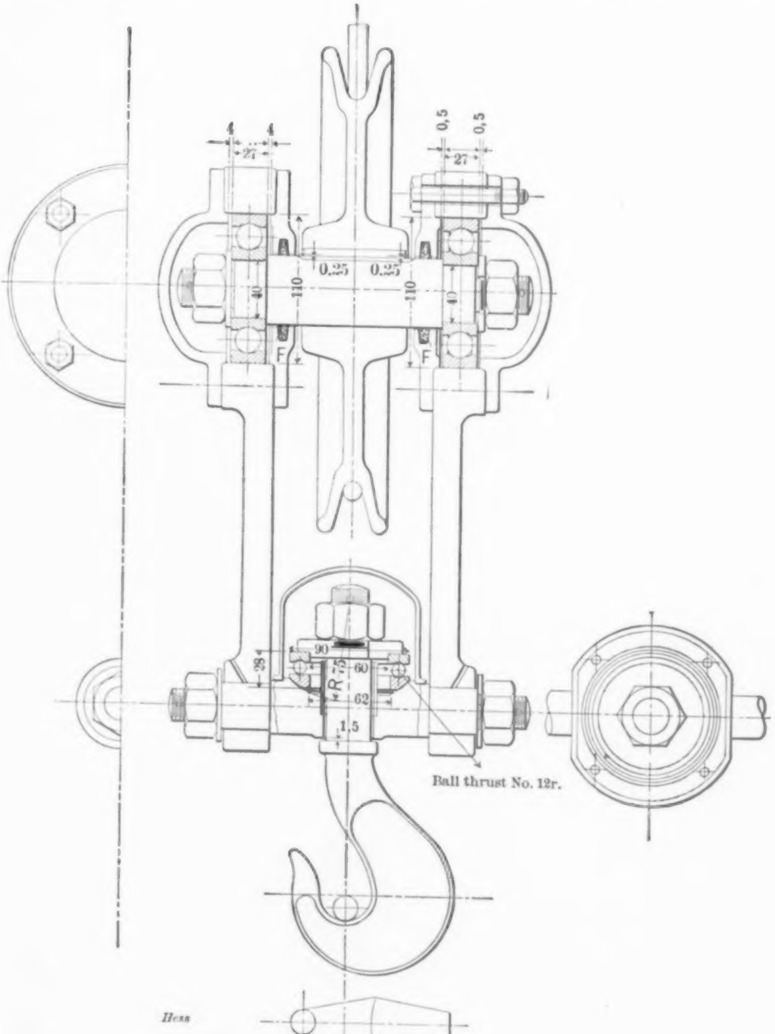


FIG. 22.—LOWER CRANE SHEAVE AND HOOK.

and 22, are all mounted on ball bearings. The half-ton load is raised at a rate of 285 feet per minute, while the slower speed of 185 feet per minute is used for the $2\frac{1}{2}$-ton load.

The mounting of this paper calender roll, Figs. 23 and 24, is the first shown in which the load is taken on two rows of

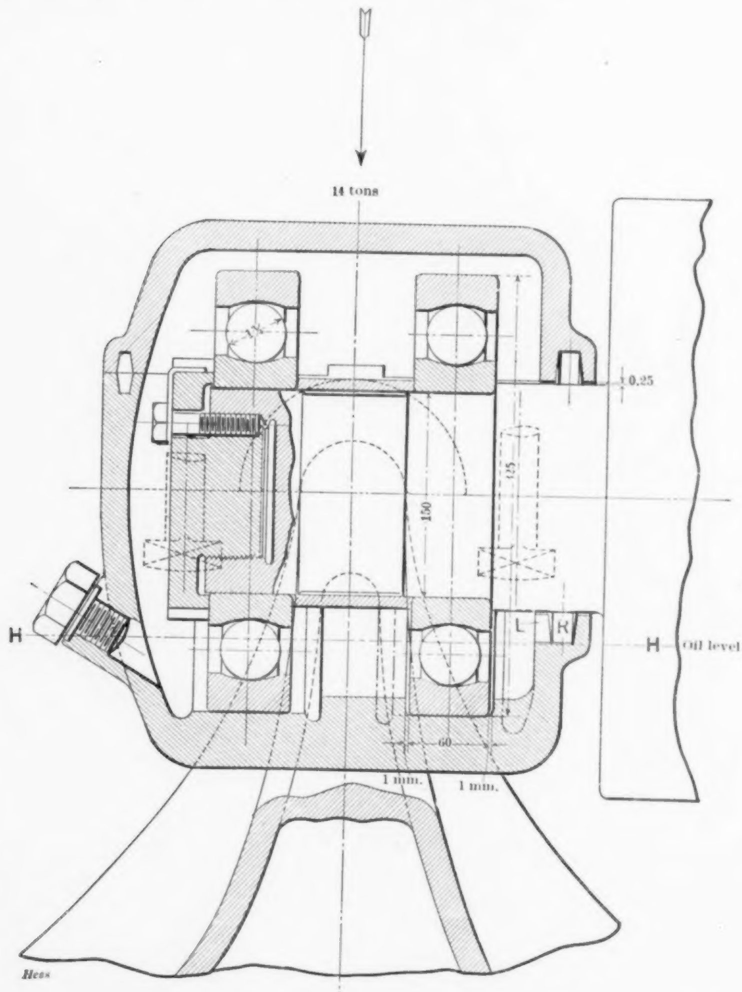


FIG. 23.—LONGITUDINAL SECTION OF PAPER CALENDER JOURNAL WITH LOAD DISTRIBUTING MOUNTING. THE INNER BALL BEARING TAKES END THRUST.

balls. The standard practice is to carry all the load on a single row of balls owing to the generally virtual impossibility of dividing it with certainty among more. Where, however, a one-sided

journal can be employed, as in railway axles, calenders, etc., an equalizer is practicable. In this case, f.i., the bearing box is hung on a trunnion (dotted lines) midway between the two rows of balls. The outer races of the ball bearings are loaded on one side only and entirely free on the other. Both will therefore receive an equal part of the load; this distribution will be unaf-

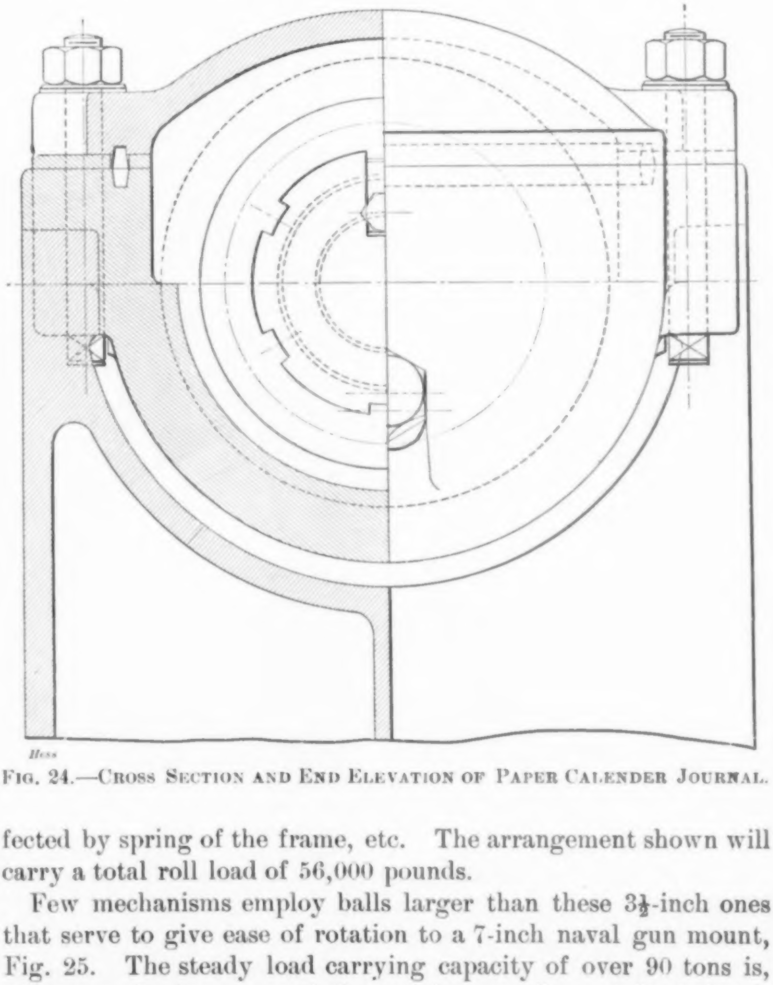


FIG. 24.—CROSS SECTION AND END ELEVATION OF PAPER CALENDER JOURNAL.

fected by spring of the frame, etc. The arrangement shown will carry a total roll load of 56,000 pounds.

Few mechanisms employ balls larger than these 3½-inch ones that serve to give ease of rotation to a 7-inch naval gun mount, Fig. 25. The steady load carrying capacity of over 90 tons is, of course, far in excess of the weight actually sustained, but is supplied to take care of the shock of firing.

In the domain of electrical engineering flywheel converters,

Figs. 26 and 27, are used in German mine hoists carrying rotating loads from 4 tons at 1200 revolutions per minute to 22 tons at 600 revolutions per minute, some on one and others on two ball rows per journal. The arrangement here shown (for 22 tons) is similar to that employed for calenders with an equalizer for two ball rows, differing in that the shaft projects through the journal and that the ball races are clamped to the unchanged

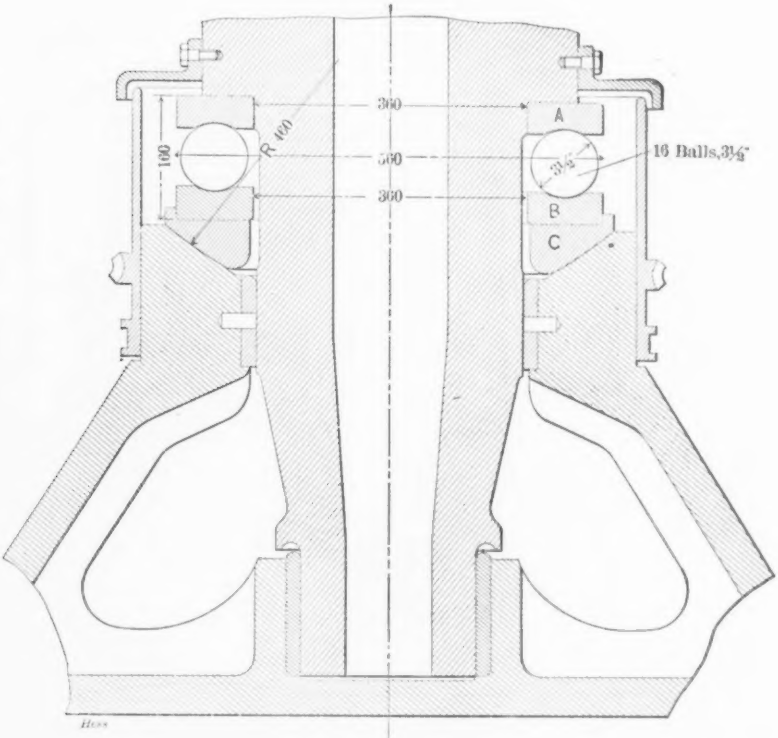


FIG. 25.—7-IN. NAVAL GUN MOUNT.

journal of the old shaft by so-called "adapters." Owing to the relatively high speed and necessity for a high degree of safety in this class of service, unusually large bearings are selected.

In the domain of transportation the railway naturally comes first. A railway box that will safely carry 9 tons per axle is shown, Figs. 28 to 31. Owing to the fact that the load is always on the top, it becomes possible to distribute this over two ball

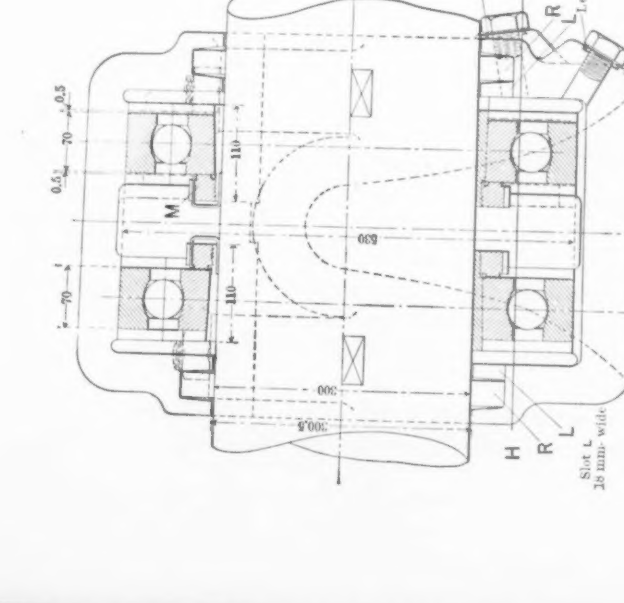


FIG. 26.

ELECTRICAL FLYWHEEL CONVERTER JOURNAL.

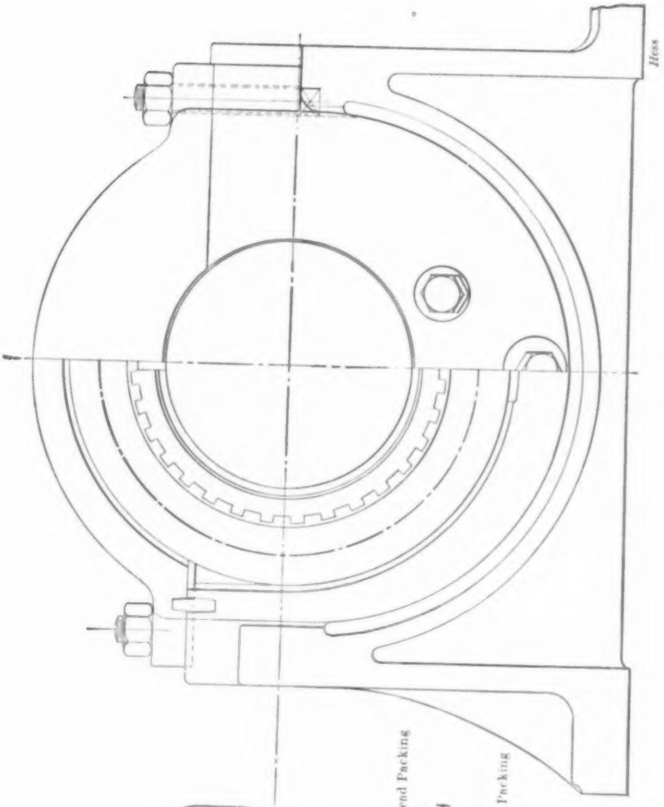


FIG. 27.

22 TONS OF ROTATING WEIGHT AT 600 R.P.M. TAKEN IN TEN JOURNALS.

bearings; in turn this permits confining the change from existing standards to the end of the axle and the box without involving the pedestal. Trials conducted by the Prussian Railway authorities up to last summer developed no appreciable wear after 110,000 miles run, and a reduction of the drawbar pull during the acceleration period of 84 per cent. and, at uniform speed, 14 per cent. on the level.

Still in the transportation field this heavy truck hub, Fig. 32, is good for a load of 5 tons on good city streets and 3½ tons for very bad rutty pavements involving much jolting.

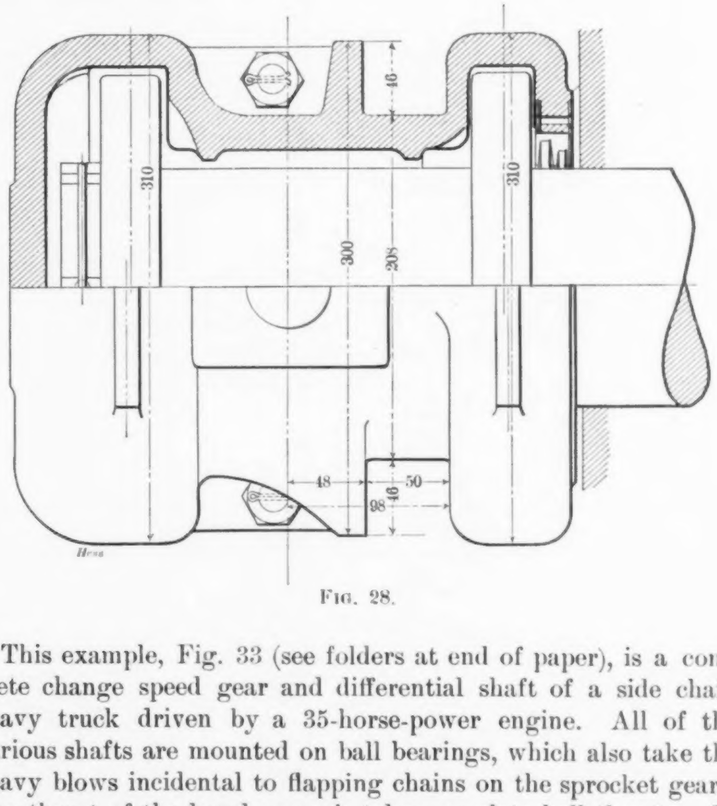


FIG. 28.

This example, Fig. 33 (see folders at end of paper), is a complete change speed gear and differential shaft of a side chain heavy truck driven by a 35-horse-power engine. All of the various shafts are mounted on ball bearings, which also take the heavy blows incidental to flapping chains on the sprocket gears. The thrust of the bevel gears is taken on plate ball thrusts. In the higher speed pleasure vehicles this thrust is usually taken on bearings of the radial type.

Possibly the most severe duty to which a ball bearing is ever subjected is in the crankshafts of gasoline engines. This

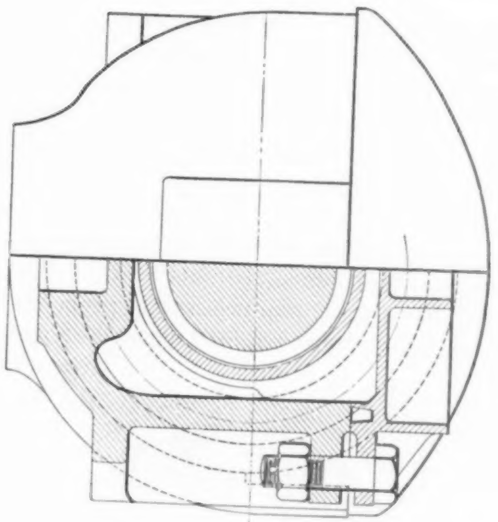


FIG. 30.

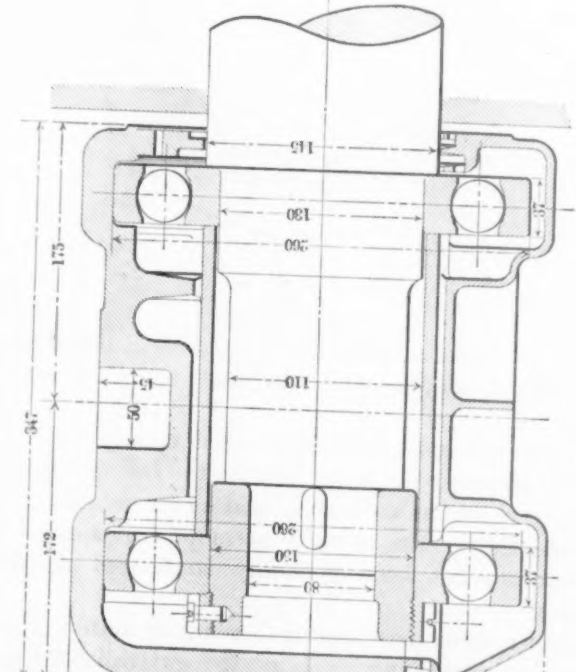


FIG. 29.

STEAM RAILWAY JOURNAL BOX. FOR 9 TONS PER AXLE LOAD.

particular example, Fig. 34 (see folders at end of paper), is from a well known French automobile, developing 20 horse-power at 900 revolutions per minute in 4 cylinders of 4-inch diameter by 5-inch stroke. Not only the main bearings, but also the crank ends of the connecting rods, are ball bearing mounted.

For the French crankshaft shown in Fig. 33 it was specified that the ball bearings should stand up under a 6-hours' test of 14 tons per journal at 450 revolutions per minute to prove their fitness for regular work under 10 tons load.

An example from naval practice is this propeller thrust block, Figs. 34 to 36, from a boat engined with 300 horse-power at 700

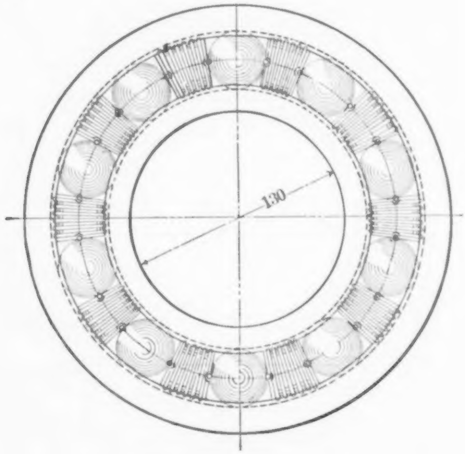


FIG. 31.

revolutions per minute. Two similar plate ball bearings take the thrust in both directions, while the shaft weight is carried in a radial bearing.

This dozen examples has been taken mainly from work of rather large dimensions, partly because the full-size drawings are more clearly visible at a distance, and again because they better demonstrate the results actually accomplished along lines that have heretofore not been generally considered as within the possibilities.

A word of caution may nevertheless not be amiss: Far greater care in proper design than with plain bearings is called for, since with these the consequences are confined to a rather more rapid wear

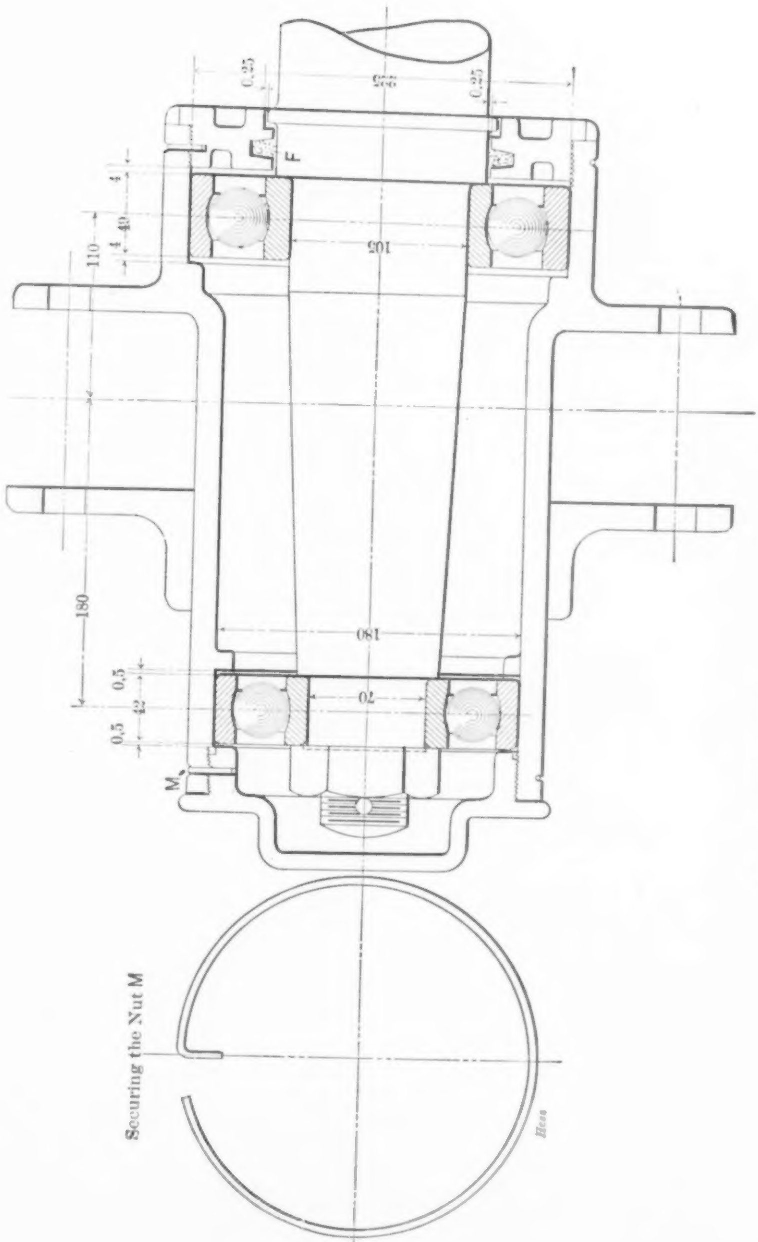


FIG. 32.—HEAVY AUTOMOBILE OR HORSE TRUCK.

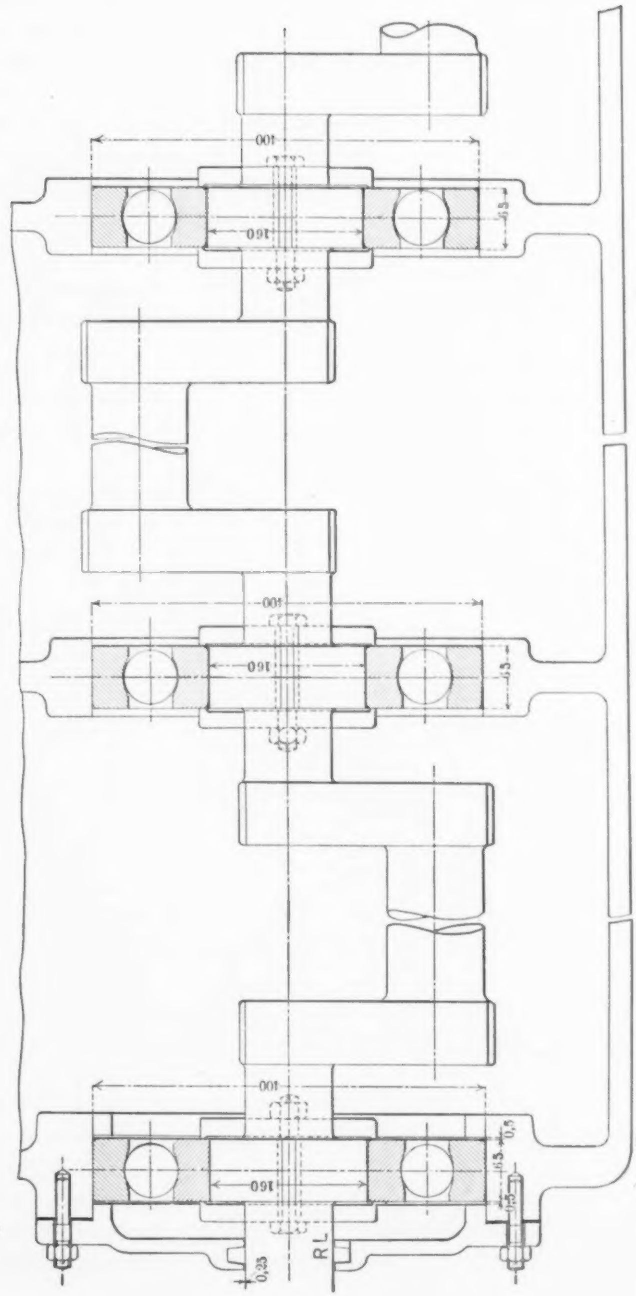


FIG. 33.—GASOLINE ENGINE CRANKSHAFT. ACCEPTANCE TEST, 14 HOURS' RUN AT 450 R.P.M. UNDER 14 TONS LOAD PER JOURNAL. REGULAR WORKING LOAD, 10 TONS PER JOURNAL AT 450 R.P.M.

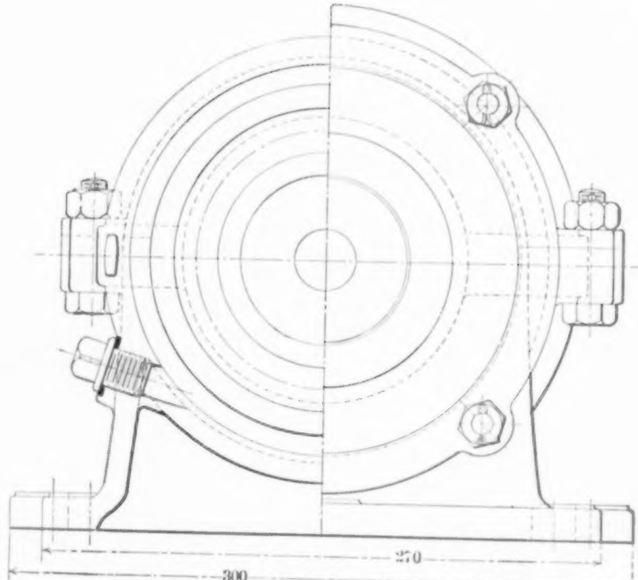


Fig. 35.

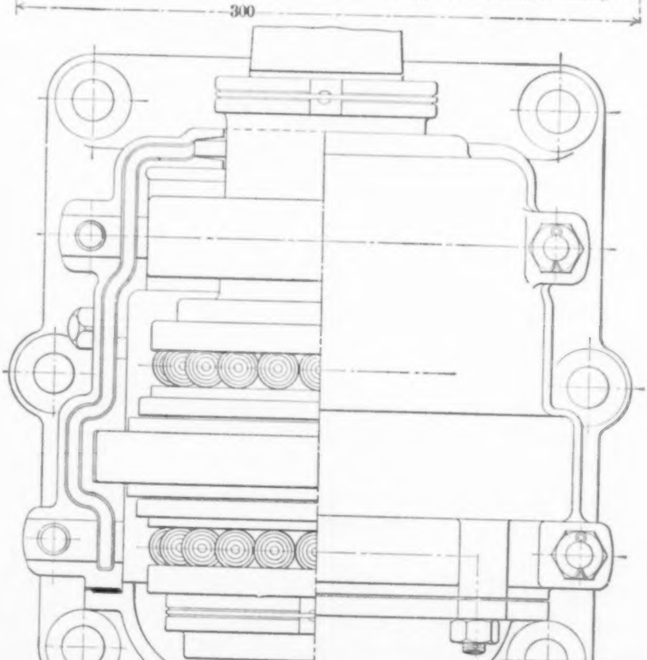


Fig. 34.

Steam Yacht Propeller Thrust and Journal Block. 300 I.H.P., 700 R.P.M.

than is desirable, while heating may be counteracted by a liberal use of cooling agents and oils. With ball bearings, however, there is no such thing as wear, but under improper conditions an actual break up of the surfaces that spells rapid destruction

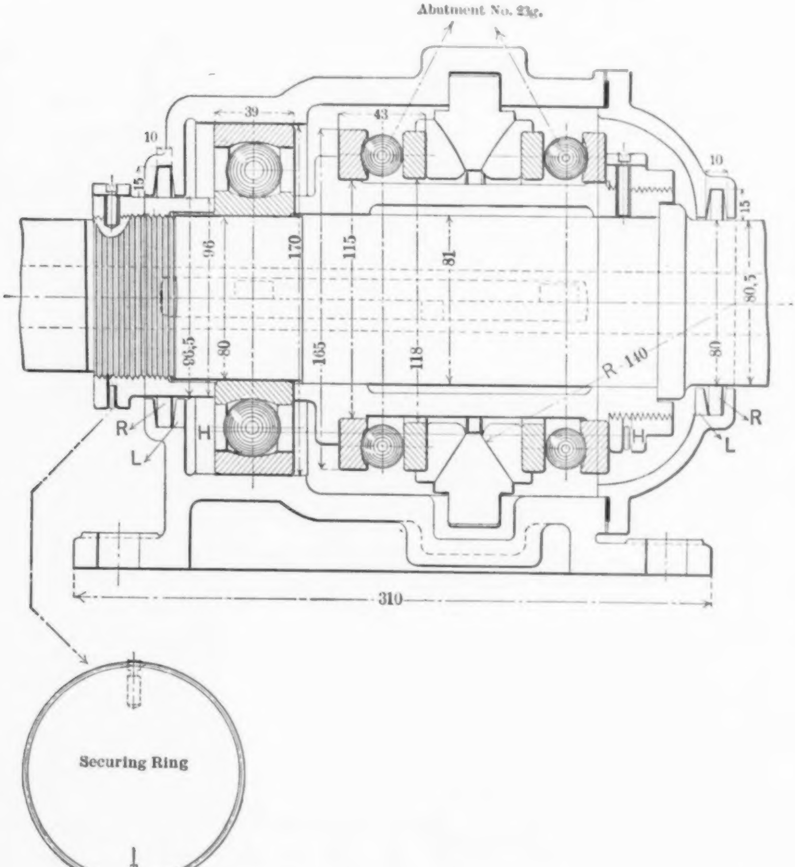


FIG. 36.—STEAM YACHT PROPELLER THRUST AND BLOCK. 300 I.H.P., 700 R.P.M.

which cannot be arrested by oiling or cooling. Then, too, the workmanship as to truth of surfaces and uniformity in size of balls must be of a very high grade and, further, the composition of the materials must be much more uniform than for other use in the mechanic arts. Finally, a definite knowledge of the

capacities of the materials employed is needed. Given these essentials, great reliability, friction elimination and other advantages result. Without these, failure and disappointment only may be looked for in repetition of the experience had before the importance of these factors was realized.

Mr. A. M. Mattice.—I will confine my remarks to a few practical observations, principally on the matters of lubrication and temperature.

In a great many cases bearings give trouble although the surface velocity and pressure per unit area are well within safe limits, the troubles being caused by the lack of attention to small but essential details.

One of the commonest causes of trouble in two-part bearings is side binding. This occurs principally in vertical engines and in other machinery where the principal load on the bearing is that due to the dead weight of the rotating part. The bearing boxes are too often bored and scraped to a good fit all around without being eased off to prevent wedging action near the parting of the boxes. This not only causes a tremendous pressure on the bearings at these points, but prevents the oil getting in between the shaft and the bearings. A bearing is always much better if eased off so that it will be well clear of the shaft for at least 20 degrees of arc on each side of the center line, and even 30 degrees is not excessive. Many designers, especially those who have not had practical experience in the operation of machinery, seem to dislike to lose any part of the bearing area by easing off the sides of the boxes, but area at these points is more detrimental than efficient, and bearings which are originally made with all around contact can frequently be improved by the use of hammer and chisel.

Give the oil a fair chance to get in its work. The edges of bearing boxes are frequently left sharp, thus scraping off the oil instead of assisting it to enter. If the box is eased off to form a channel for the oil, meeting the shaft approximately on a tangent, oil will be drawn in instead of being scraped off.

In the matter of oil grooves also, designers frequently seem loath to sacrifice bearing area, apparently losing sight of the fact that no area, no matter how great, can be sufficient unless properly lubricated. Oil grooves should be large, arranged so as to keep the oil well distributed, and should have the edges well rounded off to facilitate the entrance of oil between bearing and

journal. The simple removal of the sharp edge of the groove is not sufficient. Too many designers, moreover, seem to look upon the matter of oil grooves as of too little importance to be worthy of their consideration, but rather something to be left to the shop to take care of, or to be neglected entirely. The result frequently is an oil grooving which does more harm than good, leading the oil to certain parts of the bearing and leaving other parts dry.

From time to time various systems of improved lubrication have appeared, such as ring oiling, chain oiling, felt pads, packed waste, compression and spring grease cups, etc. But with the exception of the "splasher" system, which is very efficiently used in a number of types of inclosed engines, the old type of drop-by-drop lubrication, feeding the oil in homeopathic doses is the one most generally used with engines, although it is about the least efficient and most expensive method of all. It is not only a wasteful method, but leads to danger of cutting bearings by reducing the oil supply to a dangerous minimum. It is not altogether the fault of engine builders that this form of lubrication is the one most generally used by them, but rather because purchasers will not stand the slight additional first cost of a better method.

Ring oiling and kindred systems are very efficient for a large class of bearings, but unfortunately are not well adapted to engine work, especially in large sizes, although some designers have done some very good work along this line.

Grease and similar lubricants have their limitations, but are ideal lubricants for many purposes, and in engine work can be used to good advantage on the smaller parts of valve gear and even on eccentrics.

Supplying oil under pressure is a necessity with step bearings which carry very heavy weights, and has been applied with success to horizontal bearings where the work is extremely heavy; but this class of lubrication may be considered as adaptable only to special cases where it is an absolute necessity, to be avoided if possible.

The central gravity system of lubrication which has come into extensive use in engine installations within the past few years has resulted in a marked saving in cost of lubrication and the elimination of bearing troubles, and the details of such systems are well worth the serious attention of engineers. I refer to the systems which, while varying in details, comprise essentially an

overhead supply tank, pipes leading to all bearings with valves for regulating the supply, arrangements for catching the overflow oil, settling tanks, filters and pumps for returning the oil to the supply tank, the whole system being automatic, and resulting in a continuous circulation of oil. Although this system is at present used principally in large installations, it has been used with success in smaller plants, and I believe that engineers would be fully justified in using it in installations of single engines of large or even moderate size. Its success is not only due to the saving of labor, as in large installations, but by supplying an excess of oil to the bearings the oil does not become "worn out," as in the drop-by-drop system, resulting in a reduction of oil bills.

A further extension of the principle of continuous circulation of oil obtains in the flooded system of lubrication which has come into use in connection with steam turbines. In fact, this system was introduced by Parsons before the central gravity system for reciprocating engines came into vogue. This system consists in supplying to the bearings as much oil as will flow through them; the oil carrying away the heat of the bearings and being cooled in a tubular cooler before going back to the bearings again. The oil is not forced into the bearings under pressure, but simply supplied at a head of from a few inches to several feet; just enough to allow it to flow freely to the bearings. As the oil is nowhere exposed to the outside air, circulating only in a closed system, it collects no dirt and does not need to be filtered, but is used over and over again continuously, the entire oil supply circulating through the bearings every few minutes.

By means of this system speeds and pressures are used which would otherwise be impossible, and, what is of still greater interest to the owner, is that the oil consumption is reduced to a minimum. As instances of this I would cite the cases of two installations of 400-kilowatt steam turbines of the Parsons type running at 3600 revolutions per minute, one of which used only 50 gallons of oil in six months and the other one 55 gallons. At another plant one turbine of 400 kilowatts and another of 750 kilowatts used only three barrels of oil in sixteen months. In another case two 1000-kilowatt turbines used one-half gallon of oil per turbine week.

I crossed the Atlantic last summer in the turbine steamer "Virginian," and upon investigating the consumption of oil I found

that no oil had been added to the tanks for four successive round trips between Liverpool and Montreal, and the supply had not appreciably diminished. The service tank had a total capacity of 293 U. S. gallons, but only 144 gallons of oil had been put in, and this was being circulated through the bearings at the rate of from 40 to 50 gallons per minute, the whole supply being circulated in less than every four minutes.

The turbine steamer "Queen Alexandra," which in the summer season makes daily trips on the Clyde, used less than one barrel of oil during a season of between four and five months.

The oil consumptions of turbine steamers above quoted are for the bearings of the turbines proper, each steamer having three turbines. The line shaft bearings have ordinary lubrication, but I have no doubt that before long it will become the practice to use on such bearings the same system of flooded lubrication as on the bearings of the turbines themselves.

Users of engines and other machinery would do well to take a lesson from the results of steam turbine lubrication, which has demonstrated beyond a doubt that the supply of oil in large excess of that actually required to prevent bearings cutting is in the long run the most economical, and far in advance of the old drop-by-drop method.

Another matter upon which a few words may not be amiss is that of the temperature of bearings. There seems to be a wide misapprehension as to what is a safe temperature. I believe that much of the idea about safe temperatures is an inheritance from the time when lubricating oils were all of animal or vegetable origin and is not applicable to the high test mineral oils of to-day.

Some time ago I happened to get into a controversy as to proper bearing temperatures, the immediate cause of which was an engine whose main bearings ran at a temperature of about 135 degrees Fahrenheit, while the owner claimed that a temperature of over 100 degrees was unsafe and produced "expert" testimony to that effect. Knowing from experience that this view was not correct, but requiring testimony to the contrary, I proceeded to have examinations made of the temperature of bearings of a large number of engines of various makes. The result of this investigation showed more large engines running with bearings at temperatures over than under 135 degrees. Many bearings were running at over 150 degrees, some considerably higher, and in one case

a continuous temperature of 180 degrees was found, and in all of these cases the bearings were giving no trouble.

In this connection it might be well to point out that the introduction of direct connected electric generators has increased the temperature of engine bearings, with the same engines working under similar conditions. At first sight this may seem paradoxical, but it is easily explained. With the older types of drives the eccentrics were seldom covered, and crank oil guards, if used, were of an open type. But when driving direct connected generators, where it is necessary to keep oil off the windings and commutators, closed oil guards are fitted over cranks and eccentrics. This not only increases the temperature of the crank pins and eccentrics by preventing a circulation of air, but also retards the radiation of heat from the main bearings as well. An additional reason is that with belt or rope drive the large wheels and the belts or ropes themselves cause a considerable circulation of air, which reduces the bearing temperature, or rather the apparent temperature, as evidenced by feeling the outside of the bearing. With direct connected units the circulation of air is greatly reduced, with consequent temperature effect.

Mr. Frank Mossberg.—The Mossberg bearings are made by the Standard Machinery Company of Providence, R. I.

This company manufactures two distinct types, viz.: the cylindrical form, which is used on revolving shafts and journals to carry the radial thrust. The second form is known as an end thrust bearing, and is used to take the lateral thrust on rotating shafts that exert heavy pressure in a direction at right angles to its rotation.

We will first take up the cylindrical form of roller bearing. The underlying principle in this construction is that of distributing the load on as many rollers as possible, and with this in view the rollers are made small in diameter.

It is assumed in our practice that the rollers interposed between journal and box surface form only line contact, and consequently a small roller will carry as heavy a load as a large one.

Experiments have been made along these lines to determine diameters of rollers, and after several years' practice the following table has been adopted as our standard.

In this table the projected area of journal is adopted as the bearing surface.

MOSSBERG ROLLER BEARINGS.

Diameter of Journal. Inches.	Diameter of Rolls. Inches.	No. of Rolls.	Safe Load on Journals.
2	$\frac{1}{4}$	20	3,500
$2\frac{1}{2}$	$\frac{5}{16}$	22	7,000
3	$\frac{3}{8}$	22	13,000
4	$\frac{7}{16}$	24	24,000
5	$\frac{9}{16}$	24	37,000
6	$\frac{11}{16}$	24	50,000
7	$\frac{13}{16}$	22	70,000
8	$\frac{7}{8}$	22	90,000
9	1	24	115,000
12	$1\frac{1}{4}$	26	175,000
15	$1\frac{3}{8}$	28	255,000
18	$1\frac{3}{8}$	32	325,000
20	$1\frac{1}{2}$	34	400,000
24	$1\frac{1}{2}$	38	576,000

Surface speed of Journal from zero to 50 feet per minute.
Length of Journal $1\frac{1}{2}$ diameters.

It will be seen from this table that 20 rollers is the smallest number used in a bearing. The projected area of journal is about one third of the circumference, and consequently the smallest number of rollers supporting the load is about seven.

In rolling mill practice surface speed of journal is somewhat less, but the load very largely exceeds figures given in above table. We know of instances in actual practice where the pressure is as high as 10,000 pounds per square inch of projected area of journal.

The cage for supporting the rollers is constructed so as to obtain the most strength and best support for the rollers. The following illustration will show the construction clearly.

Material.

We find after several years' experience that the rollers should be made of tool steel not too high in carbon and should not be of too high temper. An ordinary spring temper is best. Care should be taken that the journal or shaft is not made above a medium spring temper. The box, however, should be made from high carbon steel and tempered as hard as possible.

Care in Manufacturing.

The cages are made from a good, tough bronze metal, and uttermost care is exercised to have the grooves for the rollers parallel with the axis of rotation.

The Second Form, or Thrust Bearing.

This bearing is intended to take the lateral thrust of a rotating shaft, such as a propeller shaft, worm shaft, vertical turbine shaft, gun carriage, etc., etc.

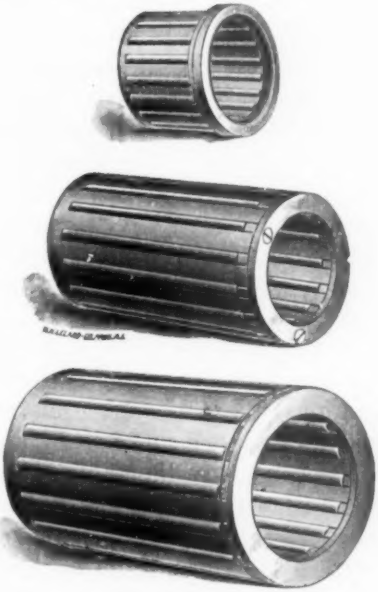


FIG. 37.

Construction.

It is obvious to any engineer that in order to insure perfect rolling contact the rollers must be conical, placed radially around the axis of rotation.

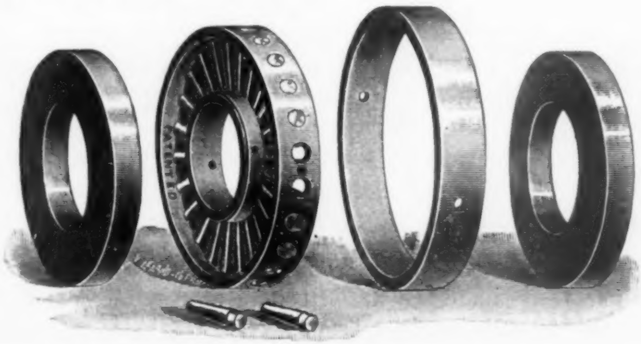


FIG. 38.

Apex of cones terminating at the center of rotation. Angles of cones not to be over from 6 to 7 degrees. If angles of rollers are larger than this, the outward thrust in radial direction will be excessive and cause destruction of outer end of rollers.

The accompanying cut shows the construction.

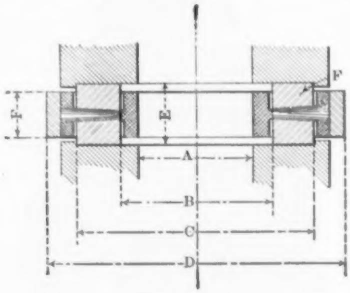


FIG. 39.

The two outer collars form the rolling surface for the bearing, one being fixed stationary and the other fastened to shaft and revolved with same. The inner surface of these two collars is made conical to correspond with the angle of rollers in bearing. The ring shown in cut fits tight over the bearing and serves to retain the rollers in position.

The collars as well as the ring are made of high carbon steel and are very hard. Rollers are made of medium carbon steel, spring temper.

Table gives principal dimension and load.

MOSSBERG THRUST BEARING.

A	D			SAFE PRESSURE ON BEARING.	
Diameter of Shaft. Inches	Inches.	No. of Rolls.	Area of Plate P.	Speed 75 Rev.	Speed 150 Rev.
$2\frac{1}{16}$—$2\frac{1}{4}$	$5\frac{9}{16}$	30	10.	19,000	9,500
$3\frac{1}{16}$—$3\frac{1}{4}$	8	30	20.	40,000	20,000
$4\frac{1}{16}$—$4\frac{1}{4}$	$10\frac{5}{16}$	30	35.	70,000	35,000
$5\frac{1}{16}$—$5\frac{1}{4}$	$12\frac{3}{8}$	30	54.	108,000	56,000
$6\frac{1}{16}$—$6\frac{1}{2}$	$14\frac{7}{8}$	30	78.	125,000	62,000
$8\frac{1}{16}$—$8\frac{1}{2}$	$18\frac{3}{4}$	32	132.	200,000	100,000
$9\frac{1}{16}$—$9\frac{1}{2}$	$20\frac{1}{2}$	32	162.	300,000	150,000

Fricttn Coefficient for the Mossberg Roller Bearings.

We do not have any very complete data on this subject, but from some tests made we find that for the cylindrical form of bearing the friction coefficient is about .002, and for the end

thrust the coefficient is somewhat larger, or about .0025. These tests have been taken by using a spring scale fastened by a cord wrapped around the journal. By pulling on the scale the shaft is started revolving. The correct method would be to measure the power required to maintain the system in motion, which is more difficult, and for this reason has not been attempted by us. We think, however, that the coefficient of friction would in such case be about 25 per cent. less.

The details as above submitted have been accumulated from practical experience of twelve years, in which time about 300 rolling mills have been put out equipped with Mossberg bearings. These mills vary in sizes from rolls 3 inches to 20 inches in diameter, mostly used for cold rolling of metals. We find that cost of maintenance is no more than with ordinary sliding bearings, and the saving of power averages more than 50 per cent. over ordinary bearings.

For paper calendars we have made bearings as large as 24 inches in diameter.

In conclusion, we wish to impress on the engineering profession our belief that roller bearings are strictly essential in modern machine construction, especially for heavy work. It is, however, a mistake for the designer to attempt the construction of such forms as may suit his own fancy. Consult the specialist in regard to what is required.

When a roller bearing is to be selected, the cheapest should not always receive the most favorable consideration. A rigid cage, with rollers carefully made to size and properly tempered, is essential.

The use of a poorly constructed roller bearing, such for instance as placing a number of rollers loosely around the journal, with merely collars at each end to keep same in position, or a roller bearing with a flimsy cage, which can easily become twisted out of shape, is bad practice, and which has led to many serious breaks and consequent general condemnation of roller bearings, as an unreliable device, which should be avoided.

On the contrary, we assert, a properly constructed and properly applied roller bearing is a most reliable device.

Mr. Samuel S. Eveland.—The use of roller and ball bearings has grown very rapidly during the past few years. In 1898 less than one hundred men were engaged in the United States in manufacturing steel balls and ball and roller bearings, while at pres-

ent over fifteen hundred are employed, and nearly as many more are in factories where applications of anti-friction bearings are the principal business, such as ball and roller bearing wagon axles, automobile axles, etc.

Notwithstanding the rapid growth of the business, it is still in its infancy, and the underlying rules governing it mechanically and from an engineering standpoint are not clearly understood. Formulas for the proper design of ball and roller bearings have been published from various sources, but they are crude and misleading, and nothing has been developed that will enable a mechanical engineer to design a satisfactory anti-friction bearing. The only reliable data on the subject have been secured by the manufacturer in actual practice, who will supply designs for bearings that will give satisfaction under specifications submitted.

Unless some essential points are correct in the application of a bearing, even one properly designed will not give satisfaction. The cup and cone and ball or roller are made true, but often they are used where the parts with which they come in contact are badly out or untrue, in consequence of which the load is forced on a few balls or rollers, making it impossible to secure satisfactory results. This accuracy in finish and alignment is the most essential point to consider in any bearing, and whether it is two, three or four point contact is of minor consideration.

Some evidence of the importance of finishing bearings accurately is shown in a test recently made by the United States government on a revolving lighthouse lens. The government engineers and mechanics designed and made in the government shops a type of ball bearing that was finished true to .001 to .002 of an inch, which was fitted with ordinary commercial balls, purchased from a jobber in New York City, which also varied in size. A manufacturer was also invited to furnish a bearing of his own design for testing in comparison with the government bearing, a ball bearing, made true both in plates and balls, so that the variation was limited to .0001 of an inch. The lens, measuring about seven feet in diameter, and weighing 3,000 pounds, was mounted on a thrust or step bearing and revolved by weighted clockwork, the rapidity of the revolutions being regulated by the amount of weight used.

The tests made and results obtained were as follows:

A. Weight required to revolve lens, with ordinary grooved

bronze and steel plate thrust and making one revolution in 30 seconds, was 900 pounds.

B. Weight required for government ball bearing, one revolution, 25 seconds, 123 pounds.

C. Weight with manufacturer's special, accurate ball bearing, one revolution in 13 seconds, 63 pounds.

D. Weight with special plates, but ordinary (government) balls, one revolution, 20 seconds, 73 pounds.

Another test was made to show the pressure required to move it, when exerted against the rim or outer circumference of the lens. In this test the government ball bearing required 64 ounces, only $\frac{1}{2}$ ounce for the special ball bearing and three ounces for the special plates and government balls.

Notwithstanding the advantages of accuracy, it is not always possible to make users take kindly to bearings accurately finished, provided it adds to the cost. In a case in point one manufacturer bought "high duty" balls, and paid an additional 10 per cent. to get extra selected balls. He later sent an order for thrust plates or washers for use with these balls, and refused absolutely to pay a trifle extra (some 15 cents each) to have the plates ground, and it was impossible to prove to him the absurdity of his position in paying extra for high grade balls and then using them on plates that varied from .002 to .003 of an inch.

Steel to Use.—Case-hardened steel gives satisfactory results when used for the plates, sleeves, cups and cones, for both ball and roller bearings. It is cheaper than tool steel, is more easily worked, and is less liable to have cracks or flaws. Another advantage is that the center, being soft, gives a cushion that very largely deadens shocks on the balls or rollers. By case-hardened steel is meant steel carbonized to a depth of $\frac{1}{32}$ to $\frac{3}{8}$ of an inch or more for heavy work. While there is some prejudice, especially abroad, to case-hardened steel, it is only because of lack of knowledge of how to treat the material satisfactorily, and which art has been developed to a high degree in the United States, so that a depth of hardening varying by $\frac{1}{64}$ of an inch can be secured at will by those who thoroughly understand the proper treatment, kind of steel to use, etc.

Oiling.—There seems to be a belief, even among engineers, that roller and ball bearings require no oiling. This is a mistake, as even if no oil was required as a lubricant, it is absolutely necessary in order to prevent rust due to moisture caused by condensation

or other causes. Aside from this, however, oil is essential to the proper running of any class of anti-friction bearing, although very little is consumed, the saving over plain bearings being from 50 per cent. to as high as 90 per cent.

Both ball and roller bearings, to give the best satisfaction, should be made of steel, hardened and ground; accurately fitted, and in proper alignment with the shaft and load; cleaned and oiled regularly, and fitted with as large size balls or rollers as possible, depending upon the revolutions per minute and load to be carried. If used as above recommended, they will return their cost many times over in power and oil saved in a very short time.

Roller Bearings.

Roller bearings are materially different from ball bearings, although generally considered practically the same by many engineers. They have some advantages in strength over ball bearings, but for high speed the latter are generally recommended.

In using roller bearings it is preferable to have the rolls of steel hardened and ground, and to run them upon hardened and ground surfaces. The rolling action of the bearing, if run on cast iron, grinds the latter into powder, and if run on soft steel, it condenses the metal, causing the bearing to become loose. Probably a few examples of results obtained by the use of roller bearings would be more interesting than a theoretical dissertation upon their advantages.

The saving in power varies, depending upon the conditions under which the bearings are used and whether they are kept cleaned, oiled and in proper alignment. On electric automobiles tests have shown that they run from 20 to 25 per cent. further on one battery charge than if plain bronze bearings are used. The United States Government tested an army wagon equipped with and without roller bearings. The result showed a saving in pull required to move the wagon on asphalt, 75 per cent.; Belgian block, 60 per cent.; common dirt road, 55 per cent.; and in starting loads on each of above, 65, 55 and 53 per cent.

An upright drill that will only drill a $\frac{5}{8}$-inch hole will drill a hole $1\frac{1}{4}$ inches in diameter when fitted with a roller thrust to take the end strain.

The steam yacht "Aphrodite," 3,400 horse-power, reports an increase in speed of 8 per cent., as well as a saving in coal of over 5 tons daily, secured by the use of roller thrust propeller bearings.

Some 30 other yachts report equally good results, as well as a great saving in oil and a reduction in vibration.

The National Tube Company have used several roller thrusts, 18-inch diameter, for five years, which show less than .001 of an inch wear, although run with a load of 100,000 pounds at 120 revolutions per minute, 16 hours daily.

A roller thrust used on a vertical motor made a test run of 24 hours with a load of 2,750 pounds at 740 to 750 revolutions per minute. It was run absolutely without oil (which was due to a misunderstanding), but the highest temperature it reached was 104 degrees; the temperature of the air in the pit where it was located was 96 degrees, showing only 8 degrees rise.

The largest anti-friction bearing ever made has recently been tested at Niagara Falls on a 5,500 horse-power unit, the rotary weight on the bearing under normal conditions being about 156,000 pounds. Under extreme conditions of head and tail water levels the load on thrust was increased by suction, etc., to a total of 190,000 pounds. The normal speed is 250 revolutions per minute, but if control is lost of the governor it may reach double that speed, or 500 revolutions per minute.

The following specifications cover the conditions under which the test was made, and which was extremely severe, so as to cover all possible contingencies.

A. With normal working pressure on balancing piston, start machine at 50 revolutions per minute; run six hours; raise speed to 75 revolutions per minute; run two hours; continue process until speed has reached 250 revolutions per minute.

B. Raise speed as rapidly as possible to 350 revolutions per minute, run five minutes, and then reduce speed as rapidly as possible to zero.

C. Raise speed quickly to 250 revolutions per minute with normal working pressure on balancing piston; run one hour to determine whether bearing is in proper running condition, after test (B). Apply load to generator in increments of 1,000-horse-power, running one hour under each load; run maximum load of 6,000-horse-power if possible.

D. Reduce pressure under balancing piston by decrements of five pounds per square inch until water is entirely cut off from under the piston, maintaining the speed at 250 revolutions per minute, and running one hour at each pressure with full load of generator.

E. Drop load suddenly, allowing the speed to rise as high as governor will permit under its normal adjustment.

The results of the test were taken every half hour for the entire period. The present thrust bearings are composed of heavy cast-iron disks, accurately faced and provided with oil grooves, running to the circumference and inclosed in tight casings, provided with plates and dead lights, through which the bearings and thermometers in the oil may be observed. Oil is furnished at about 50 pounds pressure by a set of triple acting pumps for each thrust. If by accident or otherwise the pump stops, the cast-iron thrust disks burn out at once. When this happens it involves removing thrust, casing and disks, lifting shaft, filling with new disks as well as a loss of 5,500-horse-power for at least 24 hours. The labor required for attending to the pumps amounts to about $2,000 per year, exclusive of cost of power, oil, waste, repairs, etc.

In order to lighten the duty of the thrust bearings a balance piston is filled in casing at bottom of each shaft and designed to counterbalance the entire weight of the dynamo, shafting and runner. Water is admitted to this piston at about 55 pounds per square inch and can be adjusted by valves as required. If this pressure is removed, the entire weight of at least 190,000 pounds is thrown on the thrust bearing. This may occur by shutting off the pressure through accident to the valves and piping, or, as frequently occurs in winter, by the water passages becoming clogged with ice, and when this takes place, the cast-iron thrust disks burn out at once, throwing out of commission machinery costing over $500,000 and shutting down the plant for at least two weeks. The plain roller thrust above mentioned was tried under all the above conditions of load and overload in an effort to damage or break it. In one of the tests, under conditions where the old style thrust lasted less than one minute, the roller thrust ran four hours, or 240 minutes, and showed absolutely no wear, and at the end of all the tests the bearing was in perfect condition in all respects, and less than 10 per cent. of the oil formerly used was consumed.

Steel Balls.

In making steel balls no method has been discovered to finish them all true to size. For instance, in a lot of $\frac{1}{4}$-inch balls some will measure exactly .250 of an inch; others one-quarter of a thou-

sandth large, or small, etc. In packing balls, all in each box are properly marked and of one size—.250 of an inch or O. K. size in one box, the one-quarter thousandth large in another, and one-quarter thousandth small in another, etc. This is not always understood, and the purchaser throws them all together, using them as required. In consequence, balls may vary as much as .001 of an inch or more in one bearing, which is not necessary if the trade method of packing them is understood.

In addition to the special balls, or "high duty," the grades are known as "A" or first quality and "B" or second quality. In addition to these standard grades, there are lower grades, down to culls. It is better if possible to buy direct from a manufacturer than to run the risk of getting a poor article, as it is unfortunately a fact that many hardware dealers and jobbers buy second or even lower grade balls and sell them as first grade, mixing the various sizes, etc., so that the users cannot secure proper satisfaction.

"A" grade balls vary about one-quarter thousandth of an inch. "B" grade vary about .001 to .002 of an inch, while the "high duty" or special balls are furnished varying not over one ten-thousandth of an inch.

Strength of Balls.—The crushing strength of balls should rarely be given, as it is very misleading, and has very little importance in considering the load a ball bearing will carry, as the revolutions per minute are quite as important as the load. An ordinary commercial $\frac{1}{4}$-inch ball will crush under about 6,000 to 6,500 pounds; and a $\frac{1}{2}$-inch ball 25,000 to 30,000 pounds. The same sizes will carry from 25 to 30 per cent. more in "high duty" balls and more than double that for balls made of special steel, which, however, are too expensive for general use. The fact, however, that a $\frac{1}{4}$-inch ball will crush at 6,000, 9,000 and 12,000 pounds, respectively, according to the grade, has but little bearing upon its actual strength when used commercially, as the speed is of such importance that no bearing can be designed to carry a certain weight for all purposes. Taking as a basis a bearing 3 inches in diameter, it would give approximately 9 inches the circumference of the track of the balls, requiring 12 revolutions of a $\frac{1}{4}$-inch ball to make one circumference. Half-inch balls would make it in six revolutions; $\frac{3}{4}$-inch in four revolutions, etc. Or, in assuming a speed of say 1,000 revolutions, the $\frac{1}{4}$-inch ball would have to make 12,000, $\frac{1}{2}$-inch 6,000, and $\frac{3}{4}$-inch 4,000 turns. This shows very plainly the vast importance of the revolutions

per minute. Entirely irrespective of the weight to be carried, the speed or revolutions a ball must make, if small, are so much greater than if large that it must be given first consideration, as a ball that could readily carry a load at 4,000 revolutions would be entirely unable to do so at 12,000.

Another point of importance in considering the crushing strength of a ball is the fact that the temper of the ball in hardening, at various heats, changes materially the weight it will carry. Some typewriter companies, for instance, use what is termed a "glass hard" ball, which will stand less than 50 per cent. of the strain an ordinary tempered ball will carry, and would crush instantly under even moderate shocks. They are so hard that they are literally "glass hard." At the other extreme are balls drawn at tempers especially for use where there is extreme shock or strain. Where an ordinary $\frac{1}{4}$-inch ball would crush at 6,000 pounds, the "glass hard" would do so at from 1,500 to 2,000 pounds, and the special temper at 10,000 to 12,000; yet the crushing strain they will withstand has no bearing upon the actual commercial load such a ball will carry. Therefore, great care should be exercised, in designing bearings, to use the largest balls possible, and especially for high speed.

Mr. Charles R. Pratt.—As an instance of a thrust bearing impossible to operate with friction washers, rings, balls, or cones, although all these had been tried regardless of expense, I cite my article "Roller Thrust Bearing" in the *American Machinist* of June 27th, 1901, describing a thrust bearing constructed of rolls $\frac{1}{2}$-inch in diameter by $\frac{1}{4}$-inch long (corners rounded to leave a tread of $\frac{3}{16}$ of an inch), which gave perfect results where all other known types of thrust bearings failed utterly.

In the case cited there were 180 rolls arranged in two spirals held in slots in a bronze cage. The thrust plates were 11 inches in diameter by 1 inch thick, and were of tool steel, pot hardened. The speed was 300 revolutions per minute and the thrust load was 20,000 to 80,000 pounds, which made the load per roll 111 to 444 pounds. The rolls rotated in circles of from 4 inches to 10 inches diameter.

I have found the limit of work for $\frac{1}{2}$-inch balls in thrust bearings to be: 100 pounds load per ball, at 700 revolutions per minute and with a 6-inch diameter circle of rotation.

Roller bearings at this duty are apt to be noisy.

Mr. John W. Upp.—Supplementing the topical discussion on bearings and believing it will be of interest to the Society to have the practice of one of the leading manufacturers of electrical apparatus outlined, below please find a list of clearance allowances for bearings $\frac{3}{8}$-inch to 24 inches:

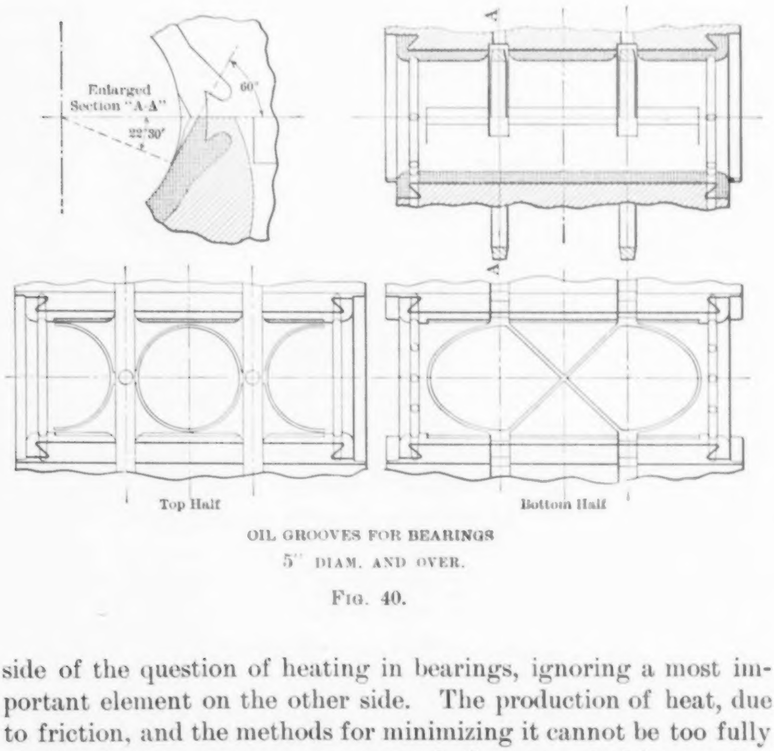
TABLE 2.

Nominal Dimension.	Journal.		Bearings.			
			Horizontal.		Vertical.	
	Maximum Diameter.	Allowable Variation below Max. Diameter.	Minimum Bore.	Allowable Variation above Min. Bore.	Minimum Bore.	Allowable Variation above Min. Bore.
	.375	.0005	377	001	376	.001
	.500	.0005	502	001	.501	.001
$\frac{5}{8}$	625	.0005	627	C01	.626	.001
$\frac{3}{4}$	.750	.0005	.752	.001	.751	.001
$\frac{7}{8}$	.875	.0005	877	.001	.876	0.1
1	1.000	0005	.002	.001	1.001	.C01
$1\frac{1}{8}$	1.125	0.05	1.128	.001	1.127	.C01
$1\frac{1}{4}$	1.250	.0005	1.253	.001	1.252	001
1	1.500	0005	1.503	001	1.502	.001
$\frac{3}{4}$	1.750	C005	1.753	.001	1.752	.00
2	2.000	.0005	2.0 3	001	2.002	.C01
$2\frac{1}{4}$	2.250	.0005	2.253	001	2.252	.001
$2\frac{1}{2}$	2.500	0005	2.503	.001	2.502	001
$2\frac{3}{4}$	2.750	0005	2. 54	002	2.753	.002
3	3.000	.C005	3.004	002	3.003	.002
$3\frac{1}{2}$	3.500	.001	3.504	002	3.503	.002
4	4.000	.001	4.005	002	4.004	002
4	4.500	.C0	4.505	002	4.504	002
5	5.000	.001	5.006	002	5.005	.002
$5\frac{1}{2}$	5.500	.001	5.5 7	2	5.506	.002
6	6.000	.001	6.009	002	6.006	.002
7	7.000	C01	.011	002	7.007	.00
8	8.000	001	8.012	.003	8.008	003
9	9.000	.0 1	9.013	004	9.009	004
10	10.000	0015	10.014	005	10.010	005
1	11.000	0015	11.015	.005	11.011	.005
12	12.000	.0015	12.016	05	12.012	.005
13	13.000	.0015	13.017	005	13.013	.C05
14	14.00	0 15	.018	005	14.014	005
15	15.000	.0015	5.019	005	15.015	.005
16	16.000	0015	16.020	005	16.016	.005
17	17.000	.0015	17.020	.005	17.016	.005
18	18.000	C015	18.020	.005	18.016	.005
19	19.000	015	19.020	.005	19.016	.C05
20	20.000	.0015	20.020	.005	20.016	.005
21	21.000	.0015	21.020	.005	21.016	.005
22	22.000	0015	22.020	005	22.016	.005
23	23.000	.0015	23.020	005	23.016	005
24	24.000	0015	24.020	005	24.016	005

I also call attention to the sketch (Fig. 40) outlining the practice of cutting oil grooves in bearings for the same class of appar-

atus, the side clearance and side channels being clearly shown. It should be noted that oil grooves are not provided for bearings 5 inches in diameter and under, but the side channels are provided on bearings of all sizes.

Mr. H. H. Suplee.—The President has made reference to the fact that future improvement in the construction of bearings must depend to a large extent upon the work of the physicist, and it appears to me that this remark is full of meaning. With but few exceptions the members who have spoken have discussed but one



OIL GROOVES FOR BEARINGS
5" DIAM. AND OVER.

FIG. 40.

side of the question of heating in bearings, ignoring a most important element on the other side. The production of heat, due to friction, and the methods for minimizing it cannot be too fully studied, but a very necessary matter is the removal of heat as fast as it is produced. Bearings become hot because the heat is allowed to accumulate in them, and their operation involves a continual interaction between the forces tending to produce heat and the opportunities offered for it to escape.

Very often the thickness of metal and its disposition about a bearing are based solely upon considerations of strength and of

resistance to the mechanical forces acting at the point under consideration. It must not be forgotten, however, that in most cases a much greater mass of metal is required to carry away the accumulating heat than is needed simply to resist the working stresses.

Bearings are sometimes cored out to permit of water circulation when the metal thus removed would have carried off the heat all right, while the removal of the metal has rendered the water circulation necessary. There is little doubt that the success which attends the use of flooded oil circulation depends as much upon the removal of the heat by the circulating oil as it does upon the improved lubrication thus effected.

It would be of the utmost value to the engineer if the physicist would conduct quantitative investigations as to the rate at which heat is carried off through the metal of a bearing, or of an engine bed or frame, under the actual conditions of engine-room service, and if tabulated data were available as to the section and mass of metal necessary to carry off a given number of thermal units per minute, it might be possible to equate this against the quantity of heat produced by the friction of the bearing, and thus determine beforehand whether a bearing would accumulate or disperse heat without waiting to find out until after it was too late. The present method suggests that of the ignorant nurse for determining the temperature of the baby's bath, the baby getting red if the water was too hot, and turning blue if it was too cold.

Mention has been made of the results with ball bearings in connection with the rotating parts of a lighthouse lamp. In this connection I may call attention to the fact that the best modern construction for such heavy vertical rotating parts involves the balancing of the weight by means of a hollow iron float in an iron tank containing mercury, the weight being thus taken almost entirely off the spindle, and the latter being required principally for guiding and holding the parts in position. A very moderate quantity of mercury is required as the float very nearly fills the tank, with but a narrow circular space for the mercury between. This arrangement is used on the powerful "feux-éclairs" or lightning flash light-houses on the coast of France, the most powerful lights in use, and I believe that a similar device is used on the light at Fire Island, so that there is no need for the bearings in such cases to support any very heavy weight.

Mr. H. G. Reist.—The pressures allowed on the projected area of journals are from 30 to 80 pounds, the average pressure being probably about 40 to 45 pounds. The lower pressure occurs in such cases where the shaft has to be enlarged on account of stiffness. The higher pressures given are generally allowed only where the rubbing speed is small. The rubbing speed of the journals for ordinary machinery varies from about 400 feet to 1200 feet per minute. The practice, as established by the General Electric Company, does not depart very far from an old formula which I understand was given by Professor Thurston many years ago, viz., the product of the pressure in pounds per square inch of projected area of the journal and the rubbing speed of the journal in feet per minute should not exceed 50,000.

The ratio of the length of bearing to the diameter generally used is 3.1, but in many places a proportionally shorter length has been found preferable.

Oil rings have been generally used by this company for many years with satisfactory results. These rings are usually placed not further than 8 inches apart. For larger bearings—a foot or more in diameter—I think it is desirable to have a circulating system of oil carrying off a considerable portion of the heat generated in the bearing by means of the oil. With the ordinary self-oiled bearing the cooling of the bearing is dependent on the radiation of the heat from the outside of the pedestal or bearing supporting case and from the shaft. No doubt a large part of the heat generated is conducted through the shaft and dissipated from the shaft some distance away from the bearing.

For self-oiled step bearings the above pressures and speeds are usually allowed. Such bearings are generally submerged in oil, and are provided with radial grooves in the moving portion of the bearing which force the oil over the bearing surface by centrifugal force. The oil may be taken up through the guide bearing and then returned to the reservoir. If the weight to be supported is of considerable size, we sometimes use a step bearing below and one above the machine, one of the bearings being rigidly connected to the shaft and the other supported from it by a spring designed to carry about half of the load.

If very great pressures have to be carried at high speeds on step bearings, it is better to support the weight on a film of oil or water maintained by pressure as is done in the now well-known "foot step bearing" used on the Curtis turbine.

In these bearings there is a circular recess, about half the total diameter of the bearing disc, to allow the oil to distribute before the shaft is raised from its seat. The pressure of the oil must be somewhat greater to raise the shaft than to support it after it is raised, since when it is in running position the pressure acts over a greater surface than the circular recess. There is a gradual fall of pressure from the recess to the outer edge where the oil is free. The distance that the bearing is raised from its seat with this arrangement is very small and depends on the rate of pumping oil through the bearing. In practice it is about .003 to .005 of an inch. The oil pressures used for such bearings vary from 250 to 800 pounds. The initial pressure while raising the bearing from its seat will be about 25 per cent. greater. A few examples of what occurs in practice may be of interest.

Weight of rotating part	9,800	53,000	187,000
Revolutions per minute	1,800	750	500
Diameter of Bearing Seat, inches	$9\frac{3}{4}$	16	$22\frac{1}{2}$
Pressure of Oil	180	420	650
Quantity of Oil in gallons per minute	1	$3\frac{1}{2}$	6

A modification of this form of bearing is used in case it is necessary to extend the shaft down through the bearing. In this case it is not possible to pump the oil in the center of the bearing, but it is forced into an annular groove in a ring forming the bearing surface. More oil is required with this arrangement, as it has two edges from which it may escape from the pressure.

Mr. H. K. Jones.—In the Russell & Erwin plant of the Corbin Screw Corporation there is in use 3,000 feet of self-oiled shafting, averaging $2\frac{7}{16}$ inches diameter, making from 150 to 500 revolutions per minute. Every bearing of this shafting has a suction oiler in the center, and some have ring oilers also at each end. So far as we have been able to discover, the bearings with suction oilers alone are working equally well with those having both kinds. There is also in the plant about 4,000 feet of shafting and 1300 separate machines which are oiled by hand.

West Virginia native oil was used exclusively for many years and did good service. In late years it has been difficult to get this oil, and for a year or two we used a black oil which is said to be a by-product in the manufacture of kerosene. This was not as good as the native oil, and between May 11th and 16th last we changed to so called "engine" oil. May 11th, with the black oil, it took 567 horse-power to drive the shop; May 16th, with

the engine oil, 487; May 24th, 472, May 31st, 454, and June 15th, 428 horse-power, being a saving of 139 out of 567 horse-power or $24\frac{1}{2}$ per cent. During the last week in July we drew out for the *first* time the old oil from our self-oiling hangers and filled them with the engine oil. I have here some of the oil taken out. It is the original oil put in thirty years ago with sufficient yearly additions to make up for evaporation. The drawing off of the old oil from the hangers had scarcely any effect on the engines. The last cards taken (November 17th) showed 427.77 horse-power with the same work being done. When we changed oil in May we simply replenished the self-oiling hangers with engine oil without draining off the old oil.

A short time ago, in a neighboring factory, I saw the caps removed from a line of hangers which had both suction and ring oilers. The oil was very thick, and nearly every one of the ring oilers was stuck fast and inoperative, but every suction oiler was in operation and abundantly oiling the shaft.

I have here one of the suction oilers, taken from a $2\frac{7}{16}$ hanger, which had no ring oilers, which has been in use 29 years and 10 months, much of the time 12 to 14 hours per day, the shaft making 300 revolutions per minute. It has worked perfectly all this time and there is a mathematical certainty that under unchanged conditions it would do so for 1,000 years.

At one time a part of our plant was run by electricity, which was furnished by three alternating current generators of 100, 150 and 200 kilowatt, 600 volt, 60 cycle, running 900, 600, 600 revolutions per minute. On more than one occasion the 100 and 200 kilowatt generator bearings were melted, although under the care of experienced electricians. From the above facts I have formed the opinion that it would be well if all generator and motor bearings were fitted with suction oilers at the center in addition to the ring oilers at the ends.

Prof. Wm. H. Kenerson.—Much has very properly been said in the discussion of ball and roller bearings regarding the desirability of careful and accurate machining and the use of balls as nearly as possible of uniform size.

Particular emphasis, however, should be laid on proper hardening since this is a matter which apparently receives too little attention on the part of the manufacturers. A familiar saying might with some truth be paraphrased to state that a ball bearing is no stronger than the weakest ball. The failure of one ball is

almost certain to cause one after the other to fail until, if not promptly discovered, the whole bearing is ruined.

From a considerable collection of examples of ball bearing failures it is evident that by far the majority were hastened, if not entirely caused, by uneven hardening, particularly of the balls.

Since it is perfectly possible by means of proper heating baths, pyrometers and methods of quenching to produce a uniform product, it seems unfortunate that high grade balls of uniform quality are so difficult to secure.

Conversation with a number of prominent manufacturers of ball bearings brings forth the opinion that the trade will not bear the additional cost of such methods of manufacture, but recent annoying and expensive experience with improperly hardened balls leads me to hope that somebody will at least try the experiment.

Mr. Geo. Hill.—Three experiences in connection with bearings may be of interest.

1. In the electrical equipment of the Auchincloss breaker for the D. L. & W. Railway, at Nanticoke, Pa.

The first coal breaker of which the writer has any knowledge in which direct current electric motors were attached directly to each part of the breaker machinery. It was feared that the dust which composes a relatively large per cent. of the atmosphere of the breaker would prove destructive to the motor bearings and commutators. As a consequence, each motor was inclosed in a galvanized iron box with locked seams, but not riveted, the opening through which the shaft projected being made one-eighth of an inch larger in diameter than the diameter of the shaft. Air was supplied to this box at about two ounces pressure through a 4-inch branch pipe from an 18-inch main pipe, into which fresh air was delivered by means of a fan from the power house located in clear air about 200 yards away from the breaker. The fresh air discharged on the motor and leaked out from the box. This arrangement has been working satisfactorily for about four years.

2. A high speed countershaft on a light automobile was very badly cut during an attempt to lubricate it properly by means of wick oil cups. The oil cups were abandoned, and over each of the two bearings a grease cup filled with graphitoleo was placed, the grease being supplied by occasionally screwing down the cups. The bearings on the shaft and the boxes were not trued

up nor touched in any way, but continued to run satisfactorily and cool during the time that the automobile remained in the writer's possession. The cutting occurred during the first thousand miles of running. The automobile was driven all told by the writer nearly 4,000 miles.

3. In the subsequent automobiles owned by the writer grease cups supplied with graphitoleo were used wherever possible, resulting in the exclusion of dust and grit from all bearings by keeping them filled with a lubricating grease. It is the writer's opinion that Albany grease would not have produced so good a result since the grease would very quickly melt and run, leaving no residue. In the case of the graphitoleo, much of the graphite remained behind, efficiently performing the functions of a lubricant.

Mr. W. S. Rogers.—I want to say, in the first place, that neither ball nor roller bearings are indispensable. They are not good everywhere for all *needs* under all conditions, and the old metallic bearing will be in use years after we have passed away.

Years ago, in coöperation with another and older member of the Society, I made experiments upon the application of such bearings to railway car journals. In our experiments we took a three-inch shaft to represent the car axle and placed a car brass of the ordinary type upon it, resisting its tendency to revolve by a weight on a string running over a small sheave.

At 100 revolutions of the shaft the car brass was pulled off and away from the wheel until the weight touched it. At 400 revolutions the friction had been reduced to such an extent that the brass returned to its normal position, and it was ready to be pulled off by the weight. Then we made a roller bearing, using three-eighths rollers.

At 100 revolutions the weight overcame the roller friction. As we increased the speed of the shaft to 450 revolutions, the roller bearing gradually followed the direction of the rotation of the shaft until the weight packed up against the sheave and the string broke. This proved clearly that the rotation of the shaft caused a friction at the higher speed with which the rollers could not keep up. Further experiments with larger rollers satisfied us that the correct design was one in which the diameter of the rollers should be the same as that of the axle.

Other inventors and experimenters knew that this was true,

since at that time rollers were patented having geared teeth, to compel the rollers to go with the shaft. The inventors since that time, or during the past ten years, have lost sight of this feature chiefly because they did not want to increase the diameter of the bearings as a whole. The effort has been to make a bearing which would go inside of the framework of the Sellers hangers for commercial reasons.

Two years ago I made an experiment abandoning the Sellers design, making a bearing of proper size and letting the hanger develop itself to suit the bearing. We put our bearings and hangers on the shafting of an old mill that had not been aligned for twenty years, with almost every foot either twisted or bent out of parallel, and in some places between hangers at eight feet centers, in a total length of sixty feet, the irregularity was equal to the diameter of the shaft. These hangers were bolted to the ceiling at angles of part of degrees, as improperly placed as possible. The shaft was driven by a water wheel whose gate, 4 feet by 12 inches, had to be wide open. Now these ball-bearings are still running and the gate is open $1\frac{1}{2}$ inches, but in my opinion the time has not come for that bearing or hanger to go on the market, since one which has not been in actual service for three to five years is an experiment as yet. In my opinion, a one-inch ball should be figured as capable of carrying not over 1,000 pounds, under maximum conditions, and a half-inch ball should not be calculated to carry over 200 pounds. Any other factor of safety less than this means trouble.

I have no use for a form of roller bearing made with small rollers with round ends, the latter bearing against an inner sleeve. The alleged advantage of such a device is the double purpose of taking both a radial load and the end thrust, but the buyer does not get two bearings in one as he thinks. The roller is pinched at one end so that its entire cage is twisted out of all semblance of parallelism or truth, and is ultimately destroyed.

Answering the point raised by Mr. Johnson, there are too many bearings made like watches—the bearings furnished to the government and used in the disappearing gun carriages are samples of this folly. The so-called silent bearing, illustrated in some of the drawings presented, is not new in this country, but has been tried off and on for many years. It is only suitable, in my opinion, for certain light loads and under mild conditions of service.

Mr. A. E. Johnson.—"Simplicity is the work of true genius,

while a complicated mechanism is a work half finished," would seem to apply to roller bearings as well as to other machines.

Probably the first bearings consisted simply of rolls inserted. Our first experience was with one of the most practical bearings ever made, the "patent sheave," in blocks. A peculiarity of these bearings is that they have so much clearance and are made of such material, that they appear to run at a lower friction co-efficient than any I ever knew of, and absolutely nothing except freezing the block solid full of ice affects them. They run on tool steel pins, which are not machined, with a clearance of about 0.06 inches, and wearing half way down through the pin apparently does not affect the operation of the bearing.

These rollers are of bronze, usually six in number, spaced nearly tangent to one another around the pin. Pin gudgeons in the ends of each roller run in circles of holes in two sheet metal rings which are loosely held by wires permitting relative rotation of perhaps ten degrees.

We have understood the O. C. R. R. equipped its Fall River trains in about '81 with a bearing which had no separators, were lubricated, and, not being perfectly protected from dust, soon clogged to delay train more than plain brasses; time gained when clean was about twenty minutes on the run. Can any member put the facts in this case on record?

One of the best of our present bearings is frequently run with but half of its rollers in a cage—why can't the other half also take care of themselves? Perhaps the manufacturer will tell us if he has made tests to determine this point. If so the results would be of value to designers.

Bearings to be used on jeweler's rolls doubtless must run without clearance or under a pressure, but a clearance of 0.004 + 0.0005 per inch diameter of journal is so little that if used in exposed positions it will soon rust solid, and the bearing is likely to be broken in small pieces if forced free without dismounting. Some things in some places should not be " fitted like a watch." The above condition can be best met by bronze rollers with the necessary increase in roller contact and clearance. In many years' experience with a simple caged bearing with about 0.05 inches clearance around the rolls in cage and 0.02 in journal diameter, we do not remember a case of one being wrecked by loosening after rusting. Cages were cast ready to receive the rolls cut from cold drawn (not rolled) steel rods, selected to vary not over 0.00025 inches.

The greatest trouble the designer has with the commercial bearings is that their makers do not give him rational formulæ by which to ascertain the relative carrying capacity. So far as we know they all come back at him with " projected area," " the larger the roll the more the bearing will carry," " steel sleeves for the journal and bushings for the bore," " rolls must not carry themselves around except through the intervening cage bar," and, worst of all, " write us, and while we are figuring it out, either sit still or take chances that you can't get our sizes in after your machine is arranged." An example of the latter case is given in which a roller bearing was designed to have a carrying capacity of 13,170 pounds and permitting a temporary increase of 60 per cent. The commercial bushing used in this place having thirteen rollers, $\frac{1}{2}$ inch diameter, 3.587 long, in the place of the fourteen rollers 4.25 long, gave but 78 per cent. of the capacity of the one the designer expected to use.

Having never experimented and become an authority, we will only present for discussion a few of the conditions we have followed during the last twenty years.

Trautwine and Christie both consider the roller capacity varies as the square root of the diameter, and as the length of roller contact, on flat surfaces. Capacity is of course greater than this in bores, because of larger arc of contact on roll and less on journals because of less arc of contact; therefore it would seem practical to design such dimensions for our slow running shafts that the bore may be of cast iron, which does not rust deeply and become pitted on its surface like steel. If bearing capacity must be increased, force a bore bushing of seamless drawn steel tube, which can be procured to 0.001 variation of diameter, into the cast iron bore and afterwards ream. The rollers running around inside will enlarge and tighten this bushing and conversely the journal bushing if used would be loosened by the rollers traveling outside of them, unless made of perhaps three times the thickness necessary for the bore bushing. A little harder steel for the shaft is sometimes better than enlarging the journal by a bushing as it will not increase the space necessary for all outside of itself—sometimes a serious matter. In arranging formulæ from common data and with the use of a constant we can compare capacity of bearings of different sizes and materials. Practically for journals you may consider that a fourth of the total length " of rolls as evenly bearing the whole load " equal to an arc of ninety degrees on the

pressure side. We have then this for our slow loads on a cast iron bore, $250\sqrt{a} \times \frac{\text{total lineal inches}}{4} = \text{capacity}$, and for say 150 feet per minute surface speed, $937.5\sqrt{d} \times \frac{\text{total lineal inches}}{4} = \text{capacity}$. The cast iron bores are by some never oiled. Steel shelves in the cast iron bores increase the above capacity 50 per cent.

The above are capacities which have been extensively used with satisfaction. Experience alone can show the proper variation of the constant for material and service, but we have from Trautwine 1,761 $\sqrt{d}$ and also statement that 2,000 $\sqrt{d}$ has been used for bridge expansions with C. R. S. rollers and untempered surfaces. One of our best manufacturers tables the capacity of his bearings, varying from 148 $\sqrt{d}$ to 1,380 $\sqrt{d}$.

Increasing the roll diameter may not increase the capacity, *e.g.*: take two 2-inch journals 4 inches long. In one case $\frac{3}{8}$ rolls 3.75 long—19 are required, capacity equaling $1250\sqrt{.375} \times \frac{19 \times 3.75}{4} = 13{,}630$ pounds load. In the other case $\frac{1}{2}$ rolls 3.75 long 14 are required, capacity equaling $1250\sqrt{.5} \times \frac{14 \times 3.75}{4} = 11600$ pounds load.

One manufacturer states that hardened and ground rollers and surfaces have borne 20,000 pounds per square inch of projected area. If he would only give the other dimensions and number of rolls we could ascertain the constant and not considering the velocity, we would then know what hardened and ground surfaces are really good for. Can anyone present this data? We believe that in a great deal of work no cage is necessary, because friction between rolls is produced by tangential pressure, and that in any case one more expensive than the one in the "patent sheave" to be worse than none. We do not understand that the tangential pressure between rolls could produce greater friction than that lost between roll and cage. The pressure is supposed to be radial and not tangential anyway.

In short there is the largest kind of a market for bushings not complicated enough to enable the manufacturer to get around another patent, but that can be sold for \$2.50, where some makers recommend one of their \$10 ones made without practical clearances.

The best analysis of capacity of balls races that we know of is

in an article by Williams in "American Machinist" of 19th of February, 1903. Has anyone applied this same analysis to the action of rollers?

Mr. Oberlin Smith.—Speaking in general of ball and roller bearings we all know they are a good thing and have come to stay. The great importance, as dwelt on here, of having them as accurately spherical or cylindrical as possible cannot be exaggerated. At the same time we must not expect too small a percentage of error, as the elasticity of the metal will take care of small variations.

I find, in talking to my assistants, that they often say, "This must not move at all; this must not bend or twist or lengthen or shorten." I reply: "Learn to consider that nearly all metals, steel and cast iron especially, behave about the same as does India rubber; only, they don't move as far, within their respective elastic limits. With metals like lead we perhaps have a nearer analogue in putty—but even lead has some resilience."

In construction, we should have our balls and rollers as round, as uniform in size and as hard as possible—that is, within reasonable commercial limits. In design we should make the rollers as long as possible, and should put enough rows of balls in a bearing so as to be sure not to crush the metal anywhere. The great thing is to keep the load far within the "yield point," and thus have balls that will last a good while, even if not throughout eternity.

Something was said about a line bearing—about it not mattering so much what was the diameter of a roller, because it was only a line bearing anyhow.

Now we speak of the point of contact of a ball or line of a roller. There is no such thing as a line contact; it is always a surface contact, with the metal more dense in the middle than at the outer edges.

Therefore, all contacts have some width, depending on the pressure, and it does make a difference whether the roller is large or small, because the larger it is the wider we get our surfaces of contact.

On a recent automobile tour a horrible noise suddenly startled us; some ball bearings gave way, and we walked home.

The trouble proved to be in the shell surrounding the balls, which was of too thin metal.

This is a common fault in such bearings. If they are too thin they may be as hard as an autoist's cheek and yet they will buckle

up and tear out sideways and do all sorts of things. If, however, they are made thick enough, and hardened to the proper temper, this trouble will not occur, and the cast is but slightly greater.

I heard something said about three or four thousand pounds to the square inch of projected area for ordinary bearings. I do not know what is generally considered a maximum, but it is well to keep it as low as possible—for durability of parts and for easy lubrication.

In a certain class of machine, the power press, it is difficult to get a low pressure per unit of surface. Presses are subject to very severe stresses, and I do not think any rule has ever been formulated as to the maximum pressure that should be put on the bearings. It is certain that this pressure is much greater than is usually allowed for other machinery.

An instance of such excessive pressure may be cited in the case of a Ferracute toggle press, where the whole ram pressure of 400 tons is brought to bear upon hardened steel toggle-pins, running in cast iron (or sometimes bronze) bearings, 3 inches in diameter by nearly 14 inches long, these having a projected area of about 40 square inches. These run habitually, for maximum work, under a load of 20,000 pounds per square inch. The case is somewhat aggravated with this particular type of press, because the ram is thrust upward from the bottom with the weight of it always keeping the toggles and pins closely in contact. These, when worn to a good fit, act almost like valves for keeping out the oil, which does not get in as plentifully as in the case of presses with toggles at the top. In these latter machines the weight of the ram and toggles pulls all the joints slightly apart, they being pushed close again when the dies come together upon the work. This makes an intermittent pressure, providing at each stroke a little clearance for oil spaces.

In a Ferracute punching press of about 84 tons capacity, which I call to mind, the pressure upon the front journal of the main shaft is about 2,400 pounds per square inch of projected area. Upon the eccentric on the front of the shaft, the pressure against the pitman driving the ram is some 7,000 pounds per square inch —both surfaces being of cast iron, and sometimes running at a surface speed of 140 feet per minute.

These pressures, when compared with ordinary practice, seem abnormally great; yet such machines run year in and year out with but little trouble in the way of heating or "cutting." The

lubrication usually consists of ordinary machine oil poured in an oil-hole once or twice a day. Of course the pressure is intermittent and continues at its maximum during only a small part of the revolution of the shaft, the amount depending on the kind of work being done.

An apparent paradox in the working of the same press is the fact that a fly wheel weighing about a ton, running freely upon the rear end of the shaft, while no work is being done, causes much more trouble in the way of getting dry and cutting than do the main journals or the eccentric, and this notwithstanding the fact that the load is less than 60 pounds pressure per square inch projected area. The reason of this doubtless is that the wheel, being slightly loose upon the stationary shaft, bears upon a so-called line of contact only, at the top of the shaft. Farther down, there is of course no steady contact. Thus the actual amount of bearing surface is vastly reduced, and the conditions are much changed from what they would be if a shaft fastened in the same wheel were running in stationary bearings of the same total area. In this latter case the shaft wears itself down and touches throughout a full half circle, as is also the case with the heavily loaded bearings previously mentioned.

Thus we have a lesson regarding the advisability of running sheaves, rollers, levers and such devices upon pins or shafts of their own, in stationary bearings, rather than to let them run loose upon a stationary pin—that is, where the pressure is heavy.

Mr. Fred. W. Taylor.—Some years ago I was intimately connected with the design of a mill for the manufacture of a new product, in which we intended originally to use 600 horse-power, but on account of intentional misrepresentation by the foreign promoters and inventors we were finally obliged to use from 1,400 to 1,700 horse-power to run the mill.

The shafting was designed for the transmission of about 600 horse-power. And this, together with the fact that, following the advice of very good authority—the latest authority at that time—we speeded many of our belts to between 5,000 and 6,000 feet per minute, led to very severe and difficult conditions for our bearings, belts, and shafting throughout the whole mill.

After three years of experience, we, of course, finally ran the mill, but in the meantime many bearings were thrown out; many proved to be inadequate to their purposes, and at the end of three years many were found to be close to the limit of durability for

proper running. Namely, many of the bearings were found to be close to the top limit of speed and pressure advisable for an ordinary, commercial mill in which the bearings do not receive or cannot be expected to receive very especial attention.

Before severing my connection with this enterprise I made up my mind, if possible, to get some benefit from the shafting experiment, and therefore I had my friend, Mr. Gulowsen, who is present, go throughout the mill and note the exact conditions of the important bearings which were running close to the limit, which were still heating a trifle and yet not dangerously heating.

On the accompanying sheet will be found the data collected in this way by Mr. Gulowsen (see Table 3 with folders at end of paper).

As a result of this data Mr. Gulowsen and the writer worked out the following formula, which we have used successfully since then, and I think it is quite trustworthy.

Let P = pressure on bearing in pounds per square inch of projected area of bearing.

Let V = velocity of circumference of bearing in feet per second.

Thus $V \times P = 400$.

The above formula is applicable to bearings in ordinary shop or mill use on shafting which is intended to run with the care and attention which such bearings usually receive, and gives the maximum or most severe duty to which it is safe to subject ordinary *chain or oiled* ball and socket bearings which are *babbitted.*

Note that it is not safe for ordinary shafting to use *cast-iron boxes,* with either sight feed, wick feed, or grease-cup oiling, under as severe conditions as the following:

$$P \times V = 200.$$

Our formula has been applied with pressures as high as 400 pounds per square inch, and accompanying velocity of 1 foot per second, on the one hand, and pressures of 40 pounds per square inch and 10 feet velocity per second, together with many intermediate combinations.

There is one other matter regarding ball bearings that I will speak of; I think it should be clearly brought out, and I do not think it has been brought out as yet. It was my good or bad fortune to be connected with a bicycle ball manufactory in Fitchburg, Mass., for some time, and many experiments were tried there upon all kinds of hardened steel balls and their bearings, and at

that time we endeavored to look up the best practice throughout the country in order to advise our customers.

I would merely emphasize this one fact that any data on ball bearings running less than two or three years of very continuous service is apt to be misleading. Any experiments extending over a few weeks are of no account whatever.

Of course, as every one familiar with the subject knows, the ball itself and the bearing gives out through fatigue at the surface of the ball or bearing. (I mean, of course, when the bearing is properly constructed. I leave out the wretched construction, and there is much of it to be found all over.) In the best construction the ball or the bearing through long use, under either too high velocities or too heavy pressure, yield through surface fatigue, which in many cases will only show itself after from three to five years of use. This fatigue shows itself first in a small speck on the ball or the bearing or both, when the surface, having been crushed or disintegrated, flakes off. And this disintegration or flaking grows and spreads until the ball or bearing goes out of use.

I want to give the warning that it is certainly not safe to draw any conclusions from any short experiments with ball or roller bearings.

Prof. A. L. Williston.—I had hoped that in connection with the discussion of this subject of bearings that the Hyatt Roller Bearing Company, of Harrison, N. J., would be able to give the Society the benefit of their experience, but unfortunately the representative whom they had promised your programme committee would be sent to represent them was unable to be present. I have since received from them, however, a considerable amount of valuable information, which I think will be of interest to the Society, and I therefore offer it as a part of the discussion of this subject.

The distinctive feature of the Hyatt Roller Bearings is the fact that it uses a *flexible* roller, which is made of a strip of steel wound into a coil or spring of uniform diameter. A roller of this construction insures flexibility, which in turn results in a uniform distribution of the load along its line of contact, both on the roller itself and on the surfaces on which it operates. It also permits any slight irregularities in either journal or box—which are always likely to occur—without causing excessive pressure. Another interesting point about this roller is that it is hollow and serves, in fact, as an oil reservoir, while the spiral interstices perform the function of carrying the oil to all parts of the bearing. The

nature of the roller, too, makes it possible to not only vary the diameter of the roller itself, but also the thickness, width and character of the stock from which the roller is made. In this way it is possible to design a bearing for any combination of conditions encountered. For instance, for a very heavy load, a roller of heavy stock can be made, while for high-speed bearing under light pressure a roller of light weight, made from thin stock, can be used. This makes it possible with rollers of the same diameter to have widely different proportions to meet the various conditions of service.

Before giving any of the results of some of the tests which have been made on these bearings I think the Society would be interested in having given a few of the points in which the makers believe that the design of their bearing is especially good.

First, the flexibility of the roller enables it to conform closely to the irregularities which are always present in the actual operation of every bearing.

Second, the flexibility of the roller permits an almost perfect distribution of the load along the entire length of the roller.

Third, the flexibility of the roller permits it to adjust itself to any slight inaccuracy in alignment between the roller and the journal, which, to a greater or lesser extent, must always occur in every bearing.

Fourth, the uniform distribution of the load which is possible in this bearing, even though the shaft is not mathematically round, permits it to be used on commercial shafting—hardened and ground journals not being essential. These bearings are suitable for operation on ordinary surfaces, except in cases of extreme conditions.

Fifth, the uniform distribution of the load on the flexible roller permits the use of ordinary soft steel surfaces, except under extreme conditions of load or steam.

Sixth, the Hyatt Bearing furnishes an ideal self-oiling bearing with a large oil reservoir in the center of every roller, which is bound to work through onto the surface of the rollers continuously.

Seventh, because of the flexibility and evenly distributed load on the roller, cutting of the surface of either shaft, roller or box is almost impossible. This greatly reduces the chances of accident to the bearing, lessens wear, and increases its durability.

Eighth, by varying not only the diameter and length of the roller, but also by varying the width, thickness and character of

stock from which the roller is made, it is possible to design a bearing which will accurately fit almost any conditions.

Ninth, the flexibility of the roller and the perfect lubrication insure a minimum coefficient friction.

Tenth, the fact that the rollers are made in automatic machines and do not have to be hardened and ground, and the fact that they may be used in connection with ordinary soft steel surfaces, reduces the cost of manufacture to a point which places it within reach of many commercial requirements where roller bearings otherwise could not be considered.

Results of Friction Tests.

A test was conducted in 1902 in one of the shops of the United Shoe Manufacturing Company of Boston. The test was conducted by the engineering department of this company for the purpose of determining the saving in this type of bearing, before deciding upon its adoption in new shops which the company was about to erect.

The line shaft which was tested was 152 feet long, 2 15-16 inches in diameter, supported by twenty bearings. It is belt-driven from one end of the shaft.

In order to obtain a fair comparison between the roller bearing and the babbitted bearings the shaft was very carefully realigned and releveled. After the first test had been made, roller bearings were then substituted, the shaft again carefully aligned and leveled, and the second test then made. The results were as follows:

Friction load of main shaft only (all countershaft belts thrown off).

Babbitted bearings...	2.28 horse-power.
Roller "	.80 "
Saving by roller bearings.	64.9 per cent.

Friction load of main shaft and 88 countershafts constantly running (countershafts in babbitted bearings).

Main shaft in babbitted bearings.	8.85 horse-power.
" " " roller "	6.36 "
Saving by roller bearings	16.7 per cent. of total.

In the same year a test was conducted in the shops of the Buffalo, Gloucester & Pittsburg Railroad at DuBois, Pa., by the officials of the Westinghouse Electrical Manufacturing Company.

The test was made to determine the friction in two-line shafts—one in a boiler shop and the other in a machine shop—both equipped with Hyatt Roller Bearings. The results are as follows:

Shaft No. 1. (Boiler shop) 170 feet long, 2¼ inches diameter, 19 hangers, speed 158 revolutions a minute, designed to transmit 30 horse-power to a 12-foot bending roll, a bolt cutter, a stay bolt cutter, a drill press, a tool grinder, a plate planer, a horizontal punch, a shear and punch, a 6-foot bending roll, a 6-foot straightening roll, and a 6-foot radial drill.

Power with all belts thrown off....................0.3 horse-power.

Shaft No. 2. (Machine shop) 180 feet long, 2½ inches diameter, 22 hangers, speed 150 revolutions a minute, designed to transmit 30 horse-power to 23 machine tools of size varying from a tool grinder to a 43-inch lathe.

Power with all belts thrown off......................0.7 horse-power.

In 1898 the Franklin Institute of the State of Pennsylvania made a series of tests to compare the friction of flexible rollers made by the Hyatt Roller Bearing Company and solid steel rollers. All the rollers were ¾ of an inch in diameter and 10 inches long. The Hyatt rollers were formed of strips of steel ½ inch wide by ⅛ inch thick. Both sets of these rollers were placed between three flat plates, and the whole set placed in a testing machine, by means of which a vertical pressure was applied. Friction between the plates and the testing machine prevented the top and the bottom plate from moving, but the middle plate was free to move on the rollers which were placed above it and below it. The horizontal force required to cause the middle plate to roll was accurately measured in each case. The results were as follows:

Total Pressure Applied.	Resistance to Motion of Plate B.	
	With Spiral Rollers.	With Solid Rollers.
2,000 pounds.	9 pounds.	26 pounds.
3,000 pounds.	17 pounds.	34 pounds.
Average	13 pounds.	30 pounds.

Increased resistance of solid rolls over spiral—131.8 per cent.

A series of tests was made at the Case School of Applied Science, of Cleveland, Ohio, to determine the coefficient of friction and a comparison of the Hyatt Roller Bearings with cast iron bearings. These tests were conducted on a friction testing machine, having an overhanging journal and a pendulum, which could be weighted, suspended from the bearings tested. The force necessary to keep the pendulum in a vertical position was measured. Speed was 480 revolutions per minute, and the diameters

of bearings tested were 1 15-16 inches, 2 13-16 inches, 2 7-16 inches, and 2 15-16 inches, in both the roller and the cast iron bearings. A summary of the results is given in the following table:

TABLE 4.

Total Weight on Bearings.	Coefficient of Friction. Roller Bearings.	Cast Iron Bearings.
Bearings $1\frac{15}{16}$ inches diameter.		
64.2	.0362	.165
114.2	.0292	.106
164.2	.0256	.116
214.2	.0218	.104
264.2	.0196	.098
Bearings $2\frac{3}{16}$ inches diameter.		
264.2	.0345	.1008
Bearings $2\frac{7}{16}$ inches diameter.		
278.8	.0292	.0765
Bearings $2\frac{15}{16}$ inches diameter.		
278.8	.0488	.0927

In examining the results of these tests it should be noted that the coefficient of friction for the roller bearings decreases rapidly in every series as the load per square inch on the bearing increases. With the cast iron bearings the coefficient of friction decreases as the load increases, but not as rapidly as in the previous case. It is fair to assume, therefore, that, if the loads on the bearings in both cases were considerably increased, the coefficient of friction would be smaller by appreciable amount than any of the figures given in the above table, and also that the difference in favor of the roller bearings would be greater. It should be noted, also, that as the size of the shaft increased in diameter the load per square inch applied was less and the coefficient of friction for the roller bearings was correspondingly increased in consequence.

A series of tests was conducted in the Laboratories of the Pride Institute, in 1904, to determine the comparative coefficient of friction of the Hyatt Roller Bearings, cast iron bearings, and bronze bearings. This series of tests was conducted to determine the advantage of using the Hyatt Rollers in the hubs of the wheels designed for a proposed incline traveling roadway. On account of the short length of the hubs of these wheels the bearings could be made but 4 inches long. The journals on which they were to run were $1\frac{1}{2}$ inches in diameter. These proportions made it im-

possible to design a bearing which would give the best results so far as the coefficient of friction was concerned; and for that reason the coefficients of friction given in the following tables are in all cases somewhat high. The conditions, however, were exactly the same for the different types of bearings. The comparison between them, therefore, is fair.

Tests were conducted at speeds varying from 128 revolutions to 585 revolutions per minute. The lubrication in all tests was with the same grade of machinery oil which was continuously fed to the bearings under a head of about one inch. This was maintained by hand. A brief summary of the results is given in the following table:

TABLE 5.

COEFFICIENT OF FRICTION.

Hyatt Roller Bearings.

Total Load.	130 R.P.M.	302 R.P.M.	585 R.P.M.
1,900 pounds.	.0114	.0090	.0181
2,700 "	.0129	.0113	.0177
3,500 "	.0124	.0109	.0164
4,300 "	.0124	.0098	.0152
5,100 "	.0115	.0097	.0135
5,900 "	.0110	.0104	.0136
6,700 "	.0105	.0096	.0128
7,500 "	.0104	.0096	.0132
8,320 "	.0101	.0094	.0124
Average	.0114	.0099	.0147

Cast Iron Bearings..

Total Load.	128 R.P.M.	302 R.P.M.	410 R.P.M.
1,900 pounds.	.0470	.0457	.0540
2,700 "	.0481	.0660	.0711
3,500 "	.0461	.0612	.0676
4,300 "	.0546	.0644	.0718
5,100 "	.0578	.0596	.0770
5,900 "	.0576	.0606	Seized.
6,700 "	.0607	.0600	"
7,500 "	.0662	.05 1	"
8,300 "	Seized.	.0561	"
Average	.0548	.0592	.0683

Bronze Bearings.

Total Load.	130 R.P.M.	320 R.P.M.	582 R.P.M.
1,100 pounds	.....	.....	.111
1,900 "	.0584	.0436	.132
2,700 "	.0567	.0552	.175
3,500 "	Seized.	.0532	Seized.
4,300 "	.....	.0784	.....
5,100 "	.....	.1002	.....
5,900 "	.....	Seized	.....
Average	.0576	.0661	.140

Another series of tests was run with a slightly smaller shaft. The shaft used in all of the tests, the results of which are given in the above table, was 1.496 inches in diameter, giving .002 inch clearance between the shaft and the boxes or rollers for oil. The second shaft was 1.494 inches in diameter, giving 50 per cent. more clearance. This smaller shaft was only used with the roller bearing, as the clearance was too much for the most satisfactory lubrication of the solid boxes.

With this shaft the average coefficient of friction, under conditions similar to those given in the above table and with the same range of loading and speeds, was .0044, or less than ½ of the coefficient friction of the larger shaft.

A series of tests was conducted at the speeds of 185 and 215 revolutions per minute, under considerably heavier loading than could be used with the solid boxes, with the results given in the following table:

TABLE 6.

HYATT ROLLER BEARINGS, COEFFICIENT OF FRICTION WITH HEAVY LOADS.

185 R.P.M.		215 R.P.M.	
Total Loads.	Coef. of Fric.	Total Loads.	Coef. of Fric.
4,000 pounds.	.0139	3,600 pounds.	.0131
7,300 "	.0131	10,800 "	.0094
12,400 "	.0106	15,200 "	.0091
17,124 "	.0089	18,200 "	.0088
19,700 "	.0086	21,900 "	.0078
....	.....	23,500 "	.0076
Average	.0110		.0093

The cast iron boxes seized the shaft at loads of from 1,000 pounds to 1,400 pounds per square inch of projected area.

The bronze boxes seized the shaft at loads of from 600 pounds to 1,000 pounds per square inch of projected area.

The Hyatt Roller Bearings, with a load of 3,942 pounds per square inch of projected area on the shaft, gave a coefficient of friction of but .0076, and continued to run smoothly without sign of heating or injury to the shaft. Tests of the starting torque were made under practically the same conditions of loading as that given in Table I. for the roller bearings, and also for the bronze boxes and for a set of babbitt-lined boxes. The average of the results of these tests is given in the following summary:

TABLE 7.

SUMMARY OF RESULTS.

	Roller Bearing Shaft 1.494 Dia.	Roller Bearing Shaft 1.496 Dia.	Cast Iron Bearing.	Bronze Bearing.	Babbitt Bearing.
Average Coefficient of Friction...........	.0044	.0118	.0608	.1120	....
Ratio...............	1.00	2.69	14.0	25.8	....
Starting Coefficient of Friction, average..	.0058	.0058	....	.091	.089
Ratio...............	1.00	1.00	....	15.7	15.3

The makers have found from their general experience that the advantages of using roller bearings of the type described are especially great when either high speeds or heavy loads are encountered. As to the formula for figuring out diameters of journals and length of boxes, it may be said that, generally speaking, the best results are obtained for line-shaft work up to speeds of 600 revolutions per minute, when a load of *30 pounds* per square inch of projected area is allowed. For conditions involving heavy load at slow speed, such as is encountered in crane and truck wheels, it has been found that a load of *500 pounds* gives the best results.

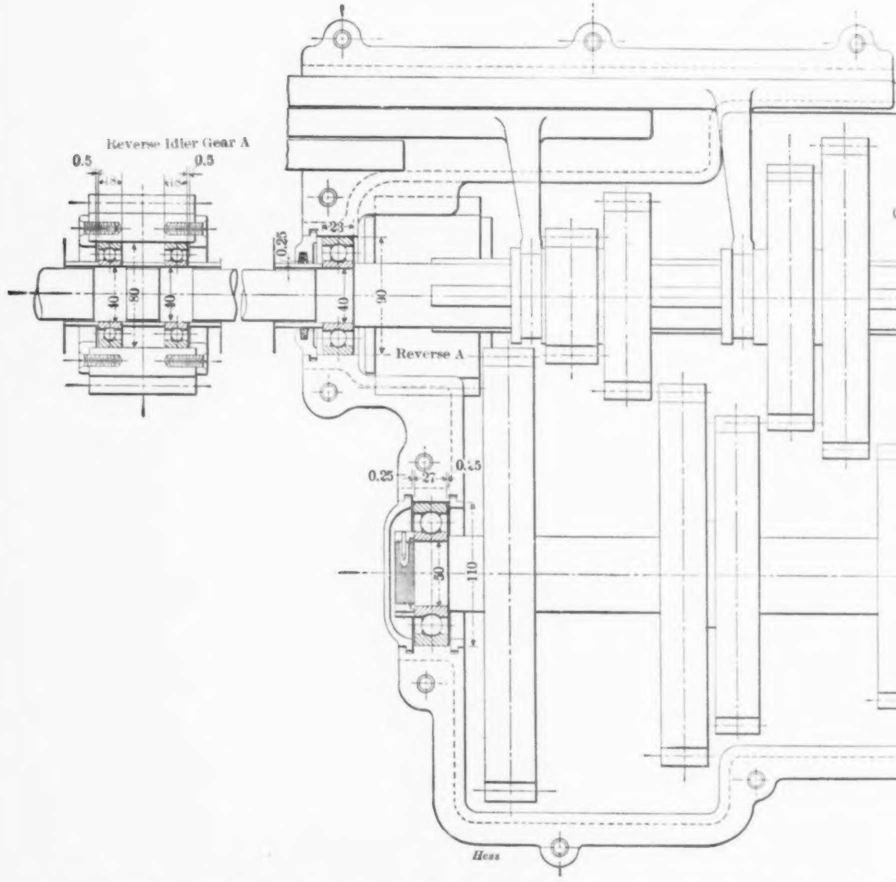
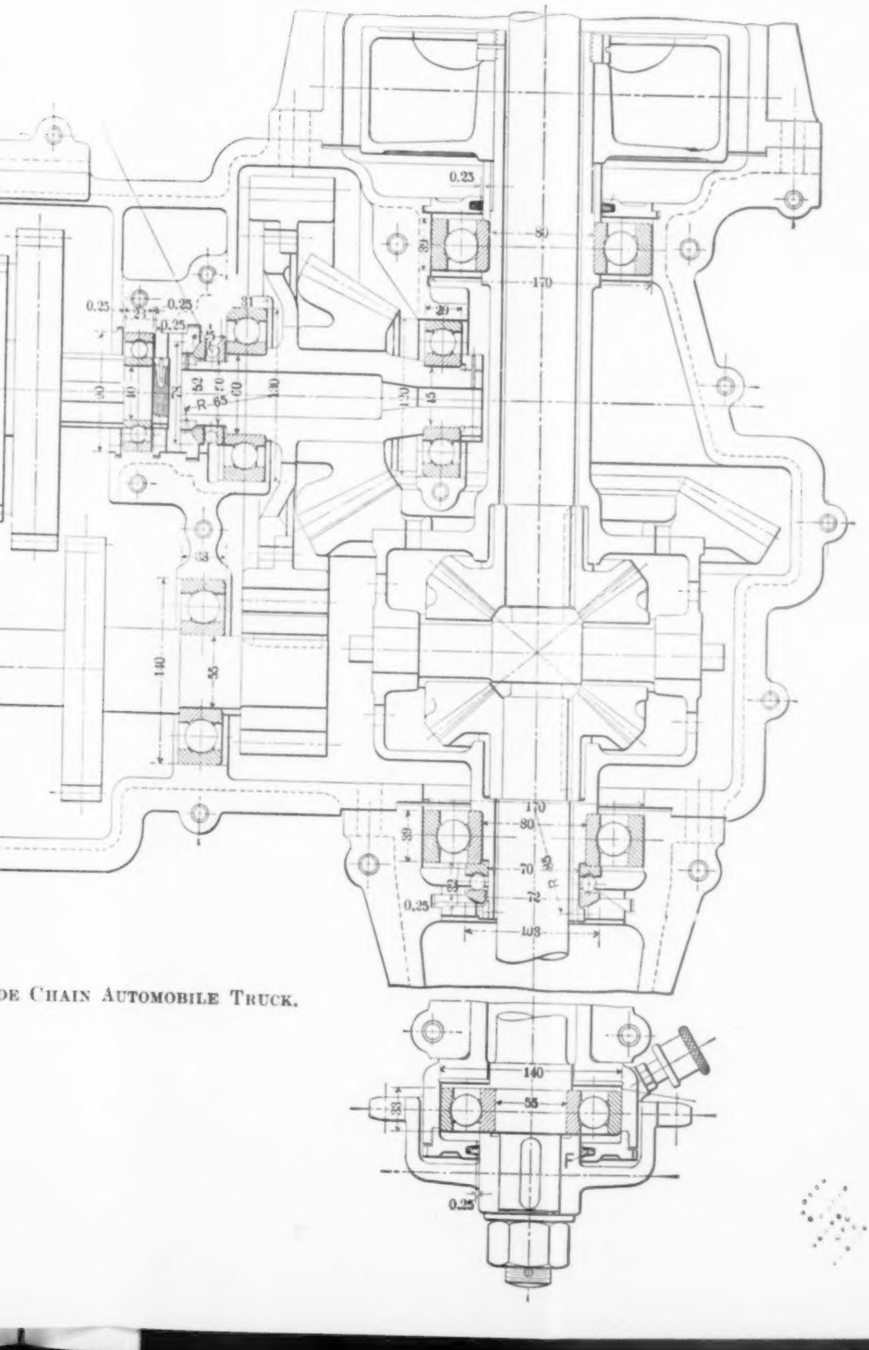


FIG. 33.—SPEED CHANGE GEAR AND DIFFERENTIAL SHAFT FOR A 35 H.P. SIDE CHA



DE CHAIN AUTOMOBILE TRUCK.

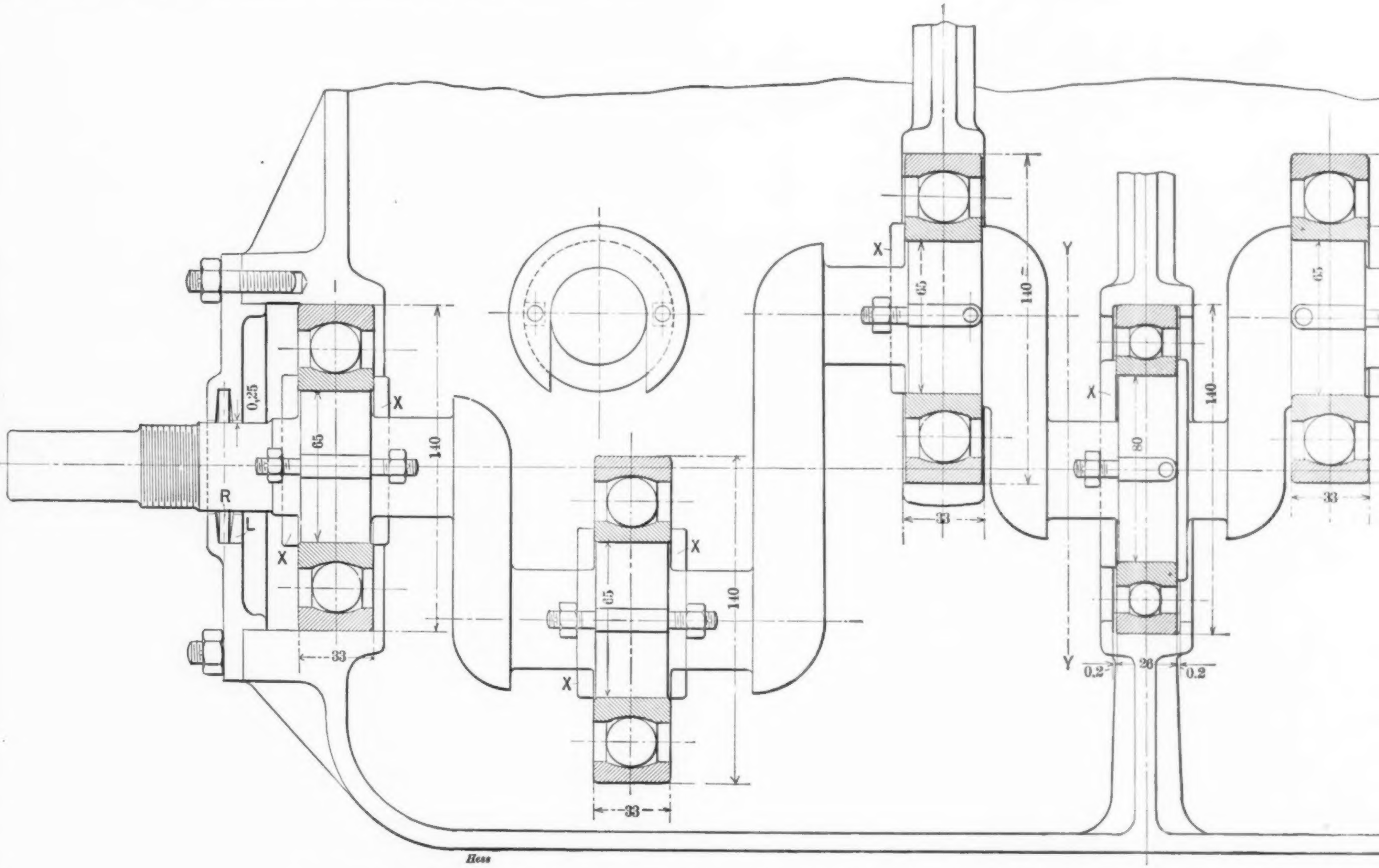
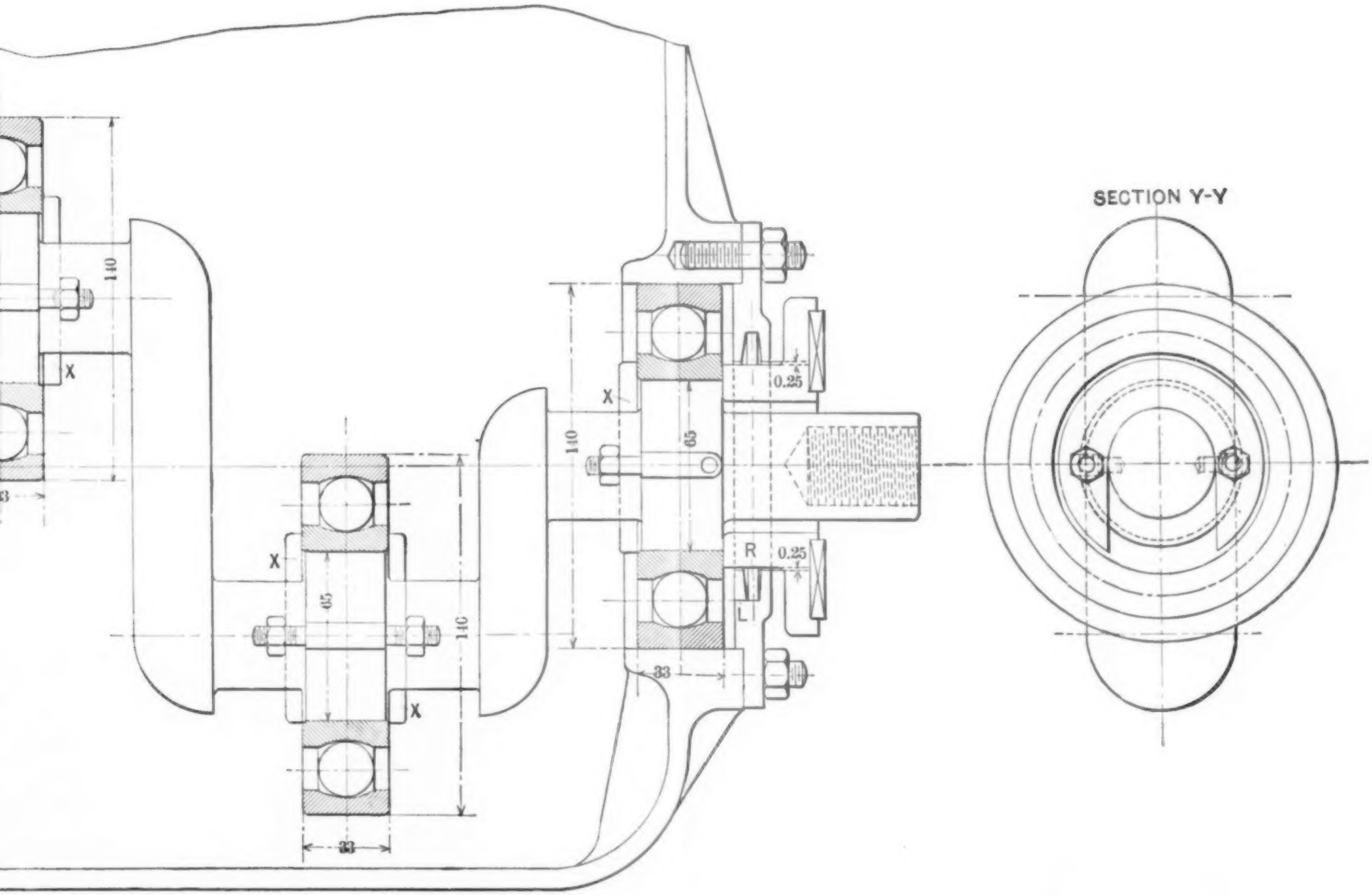


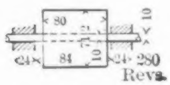
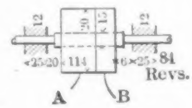
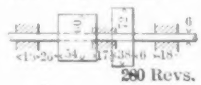
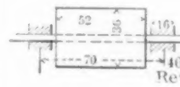
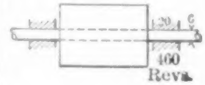
FIG. 34.—CRANKSHAFT OF A GASOLINE ENGINE AUTOMOBILE. 20 H.P. AT 900 R.P.M. 4 CYL. 4 IN. DIA. × 5 IN. STROKE.



ROKE. THE INTERMEDIATE BEARINGS HAVE SUFFICIENT BORE TO PASS OVER THE WEBS OR CRANKCHEEKS.

TABLE 3.—PRESSURE AND VELOCITY OF SHAFTS IN THEIR BEARINGS AFTER THREE YEARS

G. GULOWSEN, AU

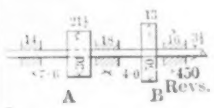
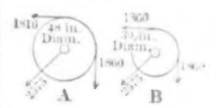
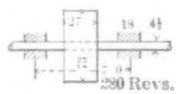
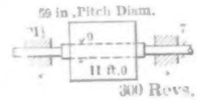
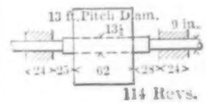
	JACK SHAFT FOR LARGE BELT DRIVE; DRIVING PUMPS AND WOOD ROOM.  1.	WHEEL SHAFT FOR LARGE BELT DRIVE.  2.	MAIN SHAFT (PUMP ROOM) CONTINUATION OF JACK SHAFT DRIVING PUMPS AND WOOD ROOM).  3.	MAIN SHAFT (WOOD ROOM) PREVIOUS TO JULY, 1892.  4.	MAIN SHAFT (WOOD ROOM) AFTER JULY, 1892.  5.	C… 6
Description of belt	70-in. × 1-in. Canvass Main Belting Co's.	70 × 1 Canvass. / 32 × $\frac{3}{4}$ Canvass.	24 × $\frac{7}{8}$ Leather / 14 × $\frac{3}{4}$ Leather	24-in. × $\frac{1}{2}$-in. Leather.	24 × $\frac{7}{8}$ Leather.	L…
Maximum tension T_1	At 230 lbs. per sq. in. 16,100	At 230 lbs. 16,100 / At 400 lbs. 9,600	at 250 lbs. 5,250 / At 200 lbs. 2,100	At 385 lbs. per. sq. in. 4,620	At 250 lbs. per sq. in. 5,250	at…
Centrifugal " T_2	7,000	7,000 / 2,400	1,450 / 1,050	820	1,450	
Available " T_3	9,100	9,100 / 7,200	3,860 / 1,050	3,800	3,800	
Slack " T_4	At 150° Contact, frict.=0.3 3,916	150° – 0.3 3,916 / 170° – 0.2 3,800	150° – 0.3 1,640 / 180° – 0.3 378	150° Contact, frict. 0.3 1,640	1,640	18…
Driving " T_5	5,184	5,184 / 3,400	2,160 / 672	2,160	2,160	
Speed of belt, ft. per. min.	5,300	5,300 / 5,300	4,400 / 5,250	4,400	4,400	
Horse-power	830, generally only 500	830 / 546	286 / 107	286	286	
Belt pull on shaft $(T_3 + T_4)$	60° downwards 13,000	60° up. 13,000 / vert. up. 11,000	5440.45° up 1428 Of these on the bearing hor. same side. 4,000 1,000	5,440, 45° downwards giving on bearing. 3,730	5,440, 45° downwards 3,730	be…
Weight of pulley and shaft	12,000	70,000	3,700 and on 3,300 2,340 bearing 1,800	2,200 1,100	2,400 1,200	
Resulting load on one bearing	12,000	27,000	4,200	4,800	4,800	
Dimensions of bearing	10 × 23	12 × 24	6 × 17	4 × 14	6 × 18	
Pressure per square inch	52.2, generally 43.	94.	41.4	86.	44.4	
Velocity, feet per second	12.2	4.4	7.3	8.	12.	
Deflection of shaft halfway between bearings	$\frac{8000.90^3}{\frac{\pi}{64}.10^4.28000000.48} = .009''$	$\frac{24000.146^3}{\frac{\pi}{64}.15^4.28000000.48} = .024'$		$\frac{4200.70^3}{\frac{\pi}{64}.4^4.28000000.48} = .085''$	$\frac{4200.70^3}{\frac{\pi}{64}.6^4.28000000.48} = .017''$	
Deflection at 6,000 lbs. per square inch	.03 in.	= .06″		= .043″	= .03″	
$T_2 = \frac{v^2.w}{32.2}$ v = velocity, feet per sec. w = weight per foot of belt. $T_3 = T_1 - T_2$. T_4 from Nystrom, 1889, page 403. $T_5 = T_3 - T_4$.	Rigid pillow blocks, babbitted. Sight feed oil cups. These bearings have given some trouble by heating, and must occasionally have a stream of water playing on shaft.	Rigid pillow blocks babbitted. Sight feed oil cups. These bearings have worked well from the start.	Ball and socket hanger babbitted boxes, large wick oil cups. This bearing required water on it periodically until a cup was used, which gave a very large flow of oil. It has run very well since.	Ball and socket pillow block babbitted boxes chain feed. This bearing gave constant trouble by heating and melting out boxes, and was finally superseded by larger one, No. 5.	Ball and socket pillow block, babbitted boxes, chain feed. This berring has run well from the start, does not heat, nor require special attention.	ba… bu… ne…

From the above data we should conclude that with a ball and socket bearing, chain or ring feed, babbitted boxes, the highest pressure and speed combination a… prefer a rather less severe duty for chain feed boxes.

It is evident from above that cast iron boxes will not stand a pressure and speed combination of 40 pounds per square inch, and 5 feet per second velocity, with…

EARS' TRIAL IN THE MANUFACTURING INVESTMENT COMPANY'S MILL AT MADISON, MAINE.

N, AUGUST 1, 1893.

OOM)	COUNTER SHAFT (WOOD ROOM) MULE BELT AND SHAVING FAN.  6.	SHAVING FAN DOUBLE FIFTY-INCH. PULLEY 12-IN. DIAM. × 13-IN., WITH BEARINGS CLOSE UP ON BOTH SIDES. 1,750 REVS. 7.	IDLER FOR MACHINE ROOM BELT IN NORTH BAY OF DIGESTER BUILDING.  8.	WALL BEARING NORTH END OF WASH ROOM.  9.	JACK SHAFT FOR LARGE ROPE DRIVE.  10.	WHEEL SHAFT FOR LARGE ROPE DRIVE.  11.
	$17\frac{1}{2} \times \frac{7}{16}$ Leather. $12 \times \frac{7}{16}$ Leather.	$12 \times \frac{7}{16}$ Leather.	$12 \times \frac{9}{16}$ Rubber.	$24 \times \frac{1}{2}$ Canvass.	24. $1\frac{3}{4}$ in. diam., four strand the Plymouth Cordage Co.	Manilla Rope furnished by
n.	at 250 lbs. At 400 lbs.	At 400 lbs. per sq. in.	At 350 lbs. per sq. in.	At 350 lbs. per sq. in.	462 lbs. per rope.	
	1,910 2,100	2,100	2,362	4,200	11,100	11,100
	506 620	620	546	1,200	202 lbs. 4,880	4,880
	1,404 1,480	1,480	1,816	3,000	260 6,220	6,220
	180° – 0.3 160° – 0.2	160° Contact, frict. = 0.2	180° Contact. – 0.2	166° Contact, frict. = 0.25.	93	
	504 835	835	948	1,450	2,220	2,220
	900 645	645	868	1,550	167 4,000	4,000
	4,300 5,800	5,800	4,770	5,400	4,650	4,650
	117 113	113	126	254	564	564
ls	$1{,}908.\frac{6}{7.5}$ $2{,}315.\frac{1}{2}$	horizontal.	45° downwards	30° downwards.	45° downwards	45° upwards
	both belts to the same side	2,315 1,157	2,575	4,450	8,440	8,440
	1,530 772					
00	$400.\frac{6}{7.5}$ $360.\frac{1}{2}$	250 125	350	2,000	9,000	30,000
	320 120					
	2,350	1,200	1,425	2,900	8,000	12,500
	$3\frac{1}{2} \times 12$	$2\frac{9}{16} \times 8$	3×11 $3\frac{1}{2} \times 12$	$4\frac{1}{2} \times 16$	7×19	9×23
	56.	58.5	43. 33.3	40	60.1	60.4
	6.9	19.9	5. 9.3	5.6	9.	4.5
7''				$\frac{5800.84^3}{\frac{\pi}{64}.4\frac{1}{2}^4.28000000.48} = .125''$	$\frac{13000.132^3}{\frac{\pi}{64}.9^4.28000000.48} = .07''$	$\frac{15000.122^3}{\frac{\pi}{64}.13\frac{1}{2}^4.28000000.48} = .01''$
''				$= .057''$	$= .072''$	$= .03''$
illow oxes, run does ecial	Ball and socket hanger babbitted boxes, chain feed. This bearing heats slightly but otherwise has given no trouble.	Regular bearings from the Sturtevant Co. gave constant trouble, being too small. They were superseded by present ones with babbitted boxes. These require a profuse supply of oil, and attention from the oiler every fifteen minutes and wear out in three months' time.	(*A*) Ball and socket hanger, cast iron grease cup. This gave constant trouble and was replaced by (*B*) Ellison.—Ball and socket babbitted boxes, chain feed. This is giving some trouble by heating.	Three sets of ball and socket pillow blocks, cast iron, grease cup wore out in a short time. Superseded by ball and socket babbitted chain feed. This has not heated or given any trouble whatever.	Ball and socket pillow block, babbitted boxes, chain feed. These have given considerable trouble from the start, melting out one box, and are overworked.	Rigid pillow blocks, babbitted boxes, chain feed. These bearings have worked well from the start.

ation admissable is 40 pounds per square inch, and 10 feet per second. This is too severe for babbitted boxes with ordinary sight feed or wick feed oiling. We should even

y, with sight feed, wick feed or grease cup oiling.

CHATTANOOGA MEETING

HELD MAY 1, 2, 3 AND 4, 1906, BEING THE
FIFTY-THIRD MEETING OF THE SOCIETY

No. 1101

PROCEEDINGS OF THE CHATTANOOGA MEETING

LOCAL EXECUTIVE COMMITTEE.

MR. NEWELL SANDERS, *Chairman*
MR. B. T. BURT, *Secretary*
MR. H. S. CHAMBERLAIN
MR. J. C. GUILD
MR. W. H. COLLIER
MR. WM. H. HUME
MAJOR H. C. NEWCOMER

The fifty-third meeting of the American Society of Mechanical Engineers was held in the city of Chattanooga, Tenn., on May 1, 2, 3, 4, 1906. The sessions for the reading of papers and the social receptions were in the meeting hall of the Masonic Temple, at Seventh and Cherry Streets. The hotel headquarters were in the Read House. Members began to arrive on Monday evening and arrived in increasing numbers on Tuesday. Occasion was taken in advance of the opening session to visit points of interest within driving distances of the city.

The opening session was called to order on the evening of Tuesday, May 1st, in the Masonic Temple, at half-past eight, by Mr. Newell Sanders, Chairman of the Local Committee.

He introduced to the meeting the Mayor of the city, the Hon. W. L. Frierson, who made an address of welcome having unusual interest and significance. His point was that each period of progress had a distinguishing mark. The world had passed through the period of military glory, the periods of excellence in

statesmanship, in art and in letters. The present day is pre-eminently the day of inventions and mechanical greatness. Recognizing in the Society the representatives of this ruling spirit of the times, the speaker expressed himself as proud to recognize the honor which the presence of the Society conferred upon the city. On behalf of the manufacturers and other citizens he bade the Society welcome.

President Fred W. Taylor, taking the chair of the meeting, replied to the welcome, giving expression to what he believed to be the sentiment of the Society, that to engineers the South was a land of promise. After the crushing poverty of forty years, and its lessons of economy and thrift, had risen the courage, determination, intelligence and character which had been the inheritance of the Southern people, and from which engineers were looking for a large industrial development.

After notice from the Secretary concerning the conduct of the meeting, a recess was ordered and an informal reception was held by the Mayor and the President.

Second Session. Wednesday, May 2.

The second session was called to order at 9.30 on the morning of Wednesday, with President Taylor in the chair. The Secretary presented the report of the Tellers under the provisions of the By-Laws, as follows:

REPORT OF TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as Tellers under By-Laws 6, 7, and 8, to scrutinize and count the ballots cast for and against the candidates proposed for membership, in their several grades, in the American Society of Mechanical Engineers and seeking election before the Chattanooga Meeting.

They met upon the designated day at the office of the Society and proceeded to the discharge of their duty. They would certify for formal insertion in the records of the Society to the election of the persons whose names appear in their several grades on the appended list.

There were 673 votes cast on the ballot closing April 21st, 1906, of which 44 were thrown out on account of informalities. The

Tellers have considered a ballot as informal which was not endorsed, or where the endorsement was made by a facsimile or other stamp.

Charles E. Lucke,
Wm. H. Bristol,
D. S. Jacobus.

Members.

Alexander, M. W.
Andrews, J. R.
Barboza, A. S.
Baum, F. G.
Berna, G. E.
Blatchley, C. A.
Brewer, A.
Burlingame, Wm. B.
Campbell, G. McK.
Card, F. M.
Carlton, W. G.
Carstens, A. B.
Cheney, N.
Churchill, Chas. O.
Cummings, E. C.
Errington, F. A.
Fletcher, J. R.
Flinn, T. C.
Flory, B. P.
Freyn, H. J. K.
Gardnier, A. J., Jr.
Head, Francis
Healy, Fred'k E.
Heap, Ray D. T.
Hemstreet, Geo. P.
Hess, Henry
Hosmer, Amos G.
Hulett, F. E.
Johnson, Edw. W.
Kingsbury, Jere G.
Knisely, Edw. S.
Lane, Fred'k
Langen, George
Lent, Leon B.
Lewis, Wm. Y.
Livingston, Robt. R.
Marquina, Luis G.
Mathot, R. E.
Miller, Geo. H.
Milne, James
Munby, Ernest J.
Norris, Edson R.
Reed, Fred M.
Ricker, Wm. W.
Riley, Robt. S.
Schwartz, Carl
Smith, Cameron C.
Smith, Otto T. R.
Spruance, W. C., Jr.
Stimson, Oscar M.
Stout, R. Paul
Timmis, Walter S.
Trotz, Johan O. E.
Warman, F. C.
Weinshank, Theo.
White, Wm. M.
Wilson, Nelson C.
Wood, Geo. R.
Young, G. A.

Promotion to Full Membership.

Booraem, J. F.
Cole, Ed. S.
Dravo, Geo. P.
Fergus, Wm. L.
Goss, Ed. O.
Hayward, Elmer L.
Hobert, S. G.
Kaup, Wm. J.
Morgan, L. H.
Neuhaus, F. A. E.
Pryor, F. L.
Scott, E. F.
Stevens, Rob. C.
Widdicombe, Robt. A.
Young, J. Paul
Young, Wm. A.

Associates.

Affleck, H. W.
Allen, W. T.
Chowins, C. E.
Conard, W. R.
Gath, A. L.
Gould, M. D.
Haden, H. Y.
Harris, Hu M.
Higgins, A. W.
Hutson, H. L.
Krzyjanowsky, C. J.
Lea, H. I.
Morgan, Wm. F.
Payson, T. E.
Shaw, C. H.
Tait, G. M. S.

Promotion to Associates.

Dewolf, E. C.
Eberhardt, H. J.
Harrington, H. G.

JUNIORS.

Adams, T. D.	Gerrish, G. H.	Ohle, E. L.
Allen, G. L., Jr.	Gleason, G. H.	Parson, C. H.
Appleton, H. W.	Harman, J. J.	Penney, R. L.
Breslove, J.	Housekeeper, Wm. G.	Polson, J. A.
Brigham, L. M.	Hoy, A. Y.	Renner, R. B.
Burton, J. H.	Huttinger, W. R.	Rice, G. W.
Conlee, G. D.	Kennedy, M. E.	Smead, W. H
Crawford, C. C., Jr.	Kirk, G. E.	Smith, L. P. C.
Davoud, V. Y.	Leonard, E. L.	Stone, E. B.
Durant, A.	Lester, B.	Thomas, J. G.
Fairchilds, G. R.	Lyon, J. L.	Van Winkle, H. H.
Fletmeyer, L. H.	Menzl, L.	White, M. G.
Francisco, F. L.	Morley, R.	Wilkes, F. C. D.
Gardner, L. H.	Morrison, E. R.	Williamson, G. E.
Gates, T. P.	Moxham, E.	

No action was required upon this report and it became a matter of record for the completion of the process of election.

The Secretary reported, also for record, a change which had been ordered and approved by the Council under the provisions of Article C59, concerning amendments to the By-Laws. The present By-Law B21 reads as follows:

FINANCIAL ADMINISTRATION.

B 21. The Council at its first meeting in each fiscal year, shall consider the recommendations of the Finance Committee concerning the expenditure necessary for the work of the Society during that year. The apportioning of the work of the Society among the various Standing and other Committees shall be on a basis approved by the Council and in harmony with the Constitution and By-Laws. The appropriations approved by the Council, or so much thereof as may be required for the work of the Society, shall be expended by the various Committees of the Society, and all bills against the Society for such expenditures shall be certified by the Committee making the expenditure and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employe of the Society except upon bills duly audited by the Finance Committee, or by resolution of the Council.

The Council had directed an amendment which should make that By-Law read as follows: beginning with the words "appropriations approved" to read:

"The appropriations approved by the Council or so much thereof as may be required for the work of the Society shall be expended by the Secretary acting as Business Manager, under the direction of the various committees of the

Society, and all bills against the Society for such expenditure shall be certified by the Secretary and shall then be sent to the Finance Committee for audit. Money shall not be paid out by any officer or employe of the Society except upon vouchers duly audited by the Finance Committee."

No action was required on this, as it was reported for record and formal transmittal to the membership.

The Secretary, on behalf of the representatives of the Society upon the Committee intrusted with the representation of the Society's interest in the building for the Engineering Societies at No. 29 West Thirty-ninth Street, New York City, the gift of Mr. Andrew Carnegie to the profession, presented a report of progress respecting the construction of that building. "At the end of April the walls were up to the thirteenth floor; the floor arches were practically all in, and the steel work completed. The electrical risers were in up to the eighth floor; the heating and ventilating work to the seventh floor, and the plumbing risers were practically all in; the boilers had been set, the guides for the elevators were partly up, and the metal lathing had been started on the fifth and sixth floors; partition work had been started on the eighth floor and the work of the electrical wiring was to begin this week. Within three weeks the outside walls would be finished and within a month all partitions would be in throughout the building; the finished floors would then be started and the plaster work well along. Three stories had been erected in seven days, which was regarded as very creditable speed."

This report called for no action beyond being made a matter of record of the convention.

Two invitations were received by the Secretary, one that the city of Charlotte, N. C., should be chosen as a place for the next spring meeting, and the other on behalf of the Jamestown Exposition Company, through Mr. H. St. George Tucker, President, that the Society holds its meeting in 1907 in Norfolk, Va.

On motion these two invitations were referred to the Council with power.

This completing the business of the Secretary's docket the Chair called for new business in order at this time.

Proposed amendments to the Constitution were presented as follows:

Mr. Chas. Wallace Hunt.—It has been thought desirable that the Society should change one of the sections of its Constitution by putting in an additional clause whereby the Council should re-

ceive authority from the Constitution to appoint an Honorary Secretary. The proposition to create this office was referred to those members of the Society who as a Committee on Constitution and By-Laws prepared the present document under which the Society is at work, so that the wording of the proposed addition should be in accordance with the other parts of that instrument.

I have in my hand the wording of an amendment to present under the provisions of Article C57, which I present in writing for discussion andt o come up for formal debate concerning its favorable consideration at the annual meeting in December. The amendment is to add to the end of Section C38 the following:

"The Council may also in its discretion appoint a person of the grade of Member to be an Honorary Secretary of the Society for a term not to exceed one year, but he may be reappointed from year to year. He shall perform such duties as may be assigned to him by the Council which are in conformity with the Constitution and By-Laws, and with or without compensation as the Council may direct."

This proposed amendment is signed by myself and Messrs. Jesse M. Smith, D. S. Jacobus, George M. Basford and R. H. Soule, who were the members of the original committee on Constitution and By-Laws.

The President called for any discussion or modification which might be acceptable to the proposers, but none being presented he announced that the amendment would take the constitutional course and come up for consideration at the annual meeting.

Mr. Jesse M. Smith presented the following:

Jesse M. Smith.—I desire to offer an amendment to the Constitution looking towards the proposition to increase the scope and influence of the Society.

It has been thought desirable that among the list of Standing Committees there should be one which in constitution and appointment would be similar to the other standing committees of the Society and which should be designated as the Research Committee.

Such committee shall consist of five members, the term of one expiring at the end of each Society year, making the committee as a body a permanent one, while changing in personnel from year to year. It is the purpose of the members presenting this proposed amendment to draft a set of By-Laws for the guidance of this Committee along the line of its work which will be presented to the Council for action and adoption under the constitutional provision made in Article C59. The amendment is to add

to Article C45 after the words "House Committee" the words "Research Committee." There are many lines of engineering in which the Society ought to be able to take a significant part and at present the mechanism for doing work along the lines of research is practically lacking. This amendment is presented by Messrs. Jesse M. Smith, C. W. Hunt, G. M. Basford, D. S. Jacobus and R. H. Soule, the same members who presented the previous proposed amendment.

President Taylor asked for discussion or acceptable modification of this proposed amendment and none being presented stated that it would take its course under the provisions of the Constitution.

No other new business being presented the regular business of the programme was taken up.

The Committee on Standard Proportions for Machine Screws presented its report with appended comment in printed form. There being no members of the committee present the Secretary read the printed paper by title and also a contribution in writing by Messrs. Burlingame and Gulowsen.

It was the sense of the meeting that in view of the absence of the reporter for the Committee and others who could present the Committee's opinion on these contributions, that the whole matter be referred back to the Committee with a request that it should make a final report including its comments on all contributed material for action at the annual meeting of the Society.

The report of the Society's committee, co-operating as an advisory body with the Pennsylvania Railroad Company in conduct of tests on locomotives at the Louisiana Purchase Exposition in St. Louis in 1904, was presented by Messrs. Goss, Herr and Sague, the Society's representatives on that advisory committee. The Secretary called attention to the fact that this advisory committee had been created at the request of the railroad company and added that formal action of thanks and recognition to the Society and its committee had been received upon the completion of the work.

Mr. E. J. Bailey had contributed discussion of the report in writing. This matter was made a matter of record in the proceedings of the meeting.

Messrs. A. W. Moseley and J. L. Bacon presented a paper on the "Effect of a Blow." Professor Jacobus called attention in discussion to the requirement that the method used should be extended.

The Secretary, on behalf of Mr. James M. Dodge, past-president of the Society, presented a paper which had been read by Mr. Dodge under the title of a "History of the Introduction of a System of Shop Management" at one of the reunions of members in New York City, together with the discussion which had been elicited thereon. Mr. Henning contributed briefly in addition to the printed discussion.

Mr. R. T. Stewart presented the results of an elaborate investigation on "Collapsing Pressures of Bessemer Steel Lap-Welded Tubes."

Messrs. Donnelly, C. W. Rice and Fred W. Taylor took part in the discussion.

Prof. Wm. H. Bristol presented with illustrations and models his paper on the development of "Low Resistance Thermo-Electric Pyrometer and Compensator." Mr. George H. Barrus took part in the discussion.

The Secretary read, on behalf of Mr. George B. Willcox, a paper on a "New Liquid Measuring Apparatus."

Third Session. Thursday, May 3.

The closing session for the reading and discussion of papers was called to order at 9.30 on Thursday morning, President Fred W. Taylor in the chair.

This session had been arranged by the Committee on Meetings to be a session of discussion on the problem of Water Wheel Governing. Papers contributing treatment of this subject were presented by Messrs. M. A. Replogle, Geo. A. Buvinger, John Sturgess and George J. Henry, Jr. Papers treating on allied subjects were also assigned to this session.

It had been Mr. Replogle's wish to illustrate his paper by lantern slides, but the mechanical and electrical difficulties introduced by a change from the direct current which had been provided for, to the alternating-current which was the only one available in the meeting hall, made this effort unsatisfactory.

The paper by Mr. Thomas E. Murray entitled "The Improvement of the Tennessee River and Power Installation of the Chattanooga and Tennessee River Power Company's Plant at Hale's Bar, Tenn.," was presented by Mr. George A. Orrok in the absence of Mr. Murray. Messrs. Donnelly, H. M. Lane, Guild, Taylor, Hunt and Hutton took part in the discussion, and Major

Newcomer of the U. S. Engineers contributed matter of great interest on the navigation problem. In the course of the discussion the question was raised whether the Society had ever taken any action in any official or effective way looking towards the preservation by the United States Government of the forests of the country in the relation of the latter to the problem of water flow in the water-sheds and ameliorating the difficulties from flood.

The Secretary, on request, stated that no action had been taken by the Society as a body, but resolutions passed in a general meeting pursuant to an opinion that a more effective way of reaching legislation in cases that had interested the Society, had been by securing the effort and influence of the individual member upon the legislators, rather than by a mere transmittal of resolutions passed at a meeting. The legislator was much more likely to be influenced by well-considered presentations from individual members in whom he had confidence, than by the sort of recommendations which could so easily be secured from an open body such as the assembled convention.

The other practice had been to refer matters of this sort to the Council with a request that that body should consider the wisest course to take up the request of the meeting.

On motion, Messrs. Gantt, Rice and Lane moved and seconded the following preamble and resolution:

"*Whereas*, the Society in session assembled at its fifty-third meeting, May 3, 1906, has considered the desirability of taking some action to secure the protection of the American forests for the preservation of the mechanical water powers:

"*Resolved:* that a committee of five be appointed by the President to consider and report to the Council on the advisability of the Society taking up in a national movement for the preservation of the forests."

The President asked Mr. Chas. Wallace Hunt whether there was anything in the Constitution of the Society unfriendly to the passage of such a resolution, and the taking of the proposed action, and on being advised that the matter was entirely within the hands of the Council to do as it seemed best, the resolution was carried.

The Secretary on behalf of Mr. Wm. O. Webber presented his paper on "Efficiency Tests of Turbine Water Wheels" with appended discussion.

Under general business, at the close of professional discussion the Secretary called attention of the members to the fact that the

paper by Mr. Thomas E. Murray which had been read at this session had been presented in two forms.

A special souvenir edition had been prepared by Mr. Murray at his own expense, in addition to the contribution of the Society, in which special edition he had incorporated much valuable information concerning Chattanooga, together with carefully prepared illustrative plates. It seemed fitting that the Society should recognize this special courtesy which the members were enjoying to advantage in their use of this special edition as a guide to Chattanooga and asked that the Society should pass a triple vote of thanks to Mr. Murray. Being duly seconded this resolution was passed.

The Secretary also called attention to the fact that the Society had been indebted not only to the Local Committee for formulating the details of the enjoyable program of the meeting, but certain Chattanooga interests which had coöperated in making the visit of the Society a pleasure. He asked that the Society should pass resolutions of thanks to the Chattanooga Railways Company, which was a consolidation of trolley and railway interests, to whose courtesy the Society was indebted for privileges of free transportation. The motion was seconded and carried.

The Nashville, Chattanooga and St. Louis Railway was to be the host of the Society on its excursion on Friday, bringing the party back to Chattanooga from the work of the navigation engineers and the Power Company, and a vote of thanks should be extended to them for the courtesy of the special train which they were to provide.

The motion duly seconded was carried.

The blanket resolution recognizing the courtesies which had been secured for the Society by Mr. Newell Sanders and his associates on the Local Committee was offered and passed by a rising vote. The Secretary was directed to transmit in the name of the Society a letter which should embody the thanks and recognition of the members.

Mr. Jesse M. Smith called attention to the fact that on the Local Committee were associated members of other engineering bodies and that their association in the work of the Local Committee there had been sounded in Chattanooga that note of accord and unity among the various branches of the profession which is beginning to be more far reaching and which is the vibrating chord underneath the workings of the building of the Engineering Societies in New York City.

The Secretary called attention also to the invitation extended to visit the plant of the tunnel company working underground at the foot of Lookout Mountain, to bring the railroad through the mountain and avoid the curves and grades now necessary to go around it. A special party was provided for the morning of Friday in advance of the regular excursion for that afternoon.

EXCURSIONS.

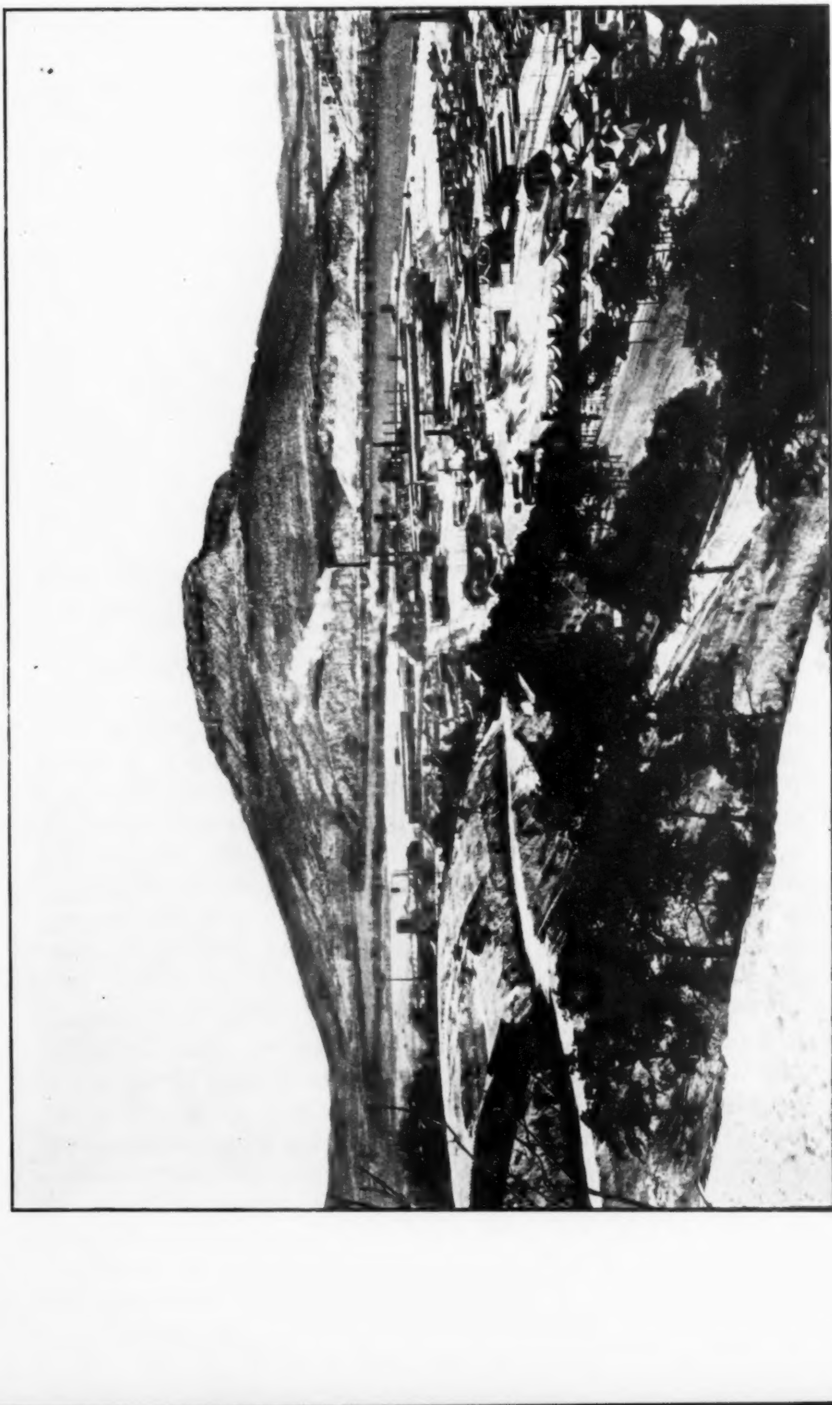
Those who attended the Chattanooga Meeting were able to participate in the hospitality of our hosts and to enjoy the excursions and entertainments which they had arranged.

The first of these was to Lookout Mountain on Wednesday afternoon, the party taking special cars from the Read House to the foot of the incline road by which the ascent to the summit was made. When the party has assembled on Lookout Point, a short but graphic account of the "Battle Above the Clouds," Missionary Ridge and the general campaign about Chattanooga was given by Hon. H. Clay Evans.

On the evening of this day the customary reception and dance was given by the Local Committee at the Masonic Hall and was well attended.

Thursday afternoon was devoted to a trip to Chickamauga Park. The party left the Read House in special cars and were met at the Park by stages and carriages, where the drive commenced. A stop was made on the drill grounds of the U. S. Army Post, where by the courtesy of the commander, Lieut.-Colonel George F. Chase, an exhibition drill was given by the 12th U. S. Cavalry. The drive was then continued along the crest of Missionary Ridge, returning to Chattanoga, and the hotel in time to permit the members to attend a concert given by Mr. Walter Damrosch. This latter was not on the program, but was greatly enjoyed by those who attended.

The last excursion was the delightful trip down the Tennessee River in the steamboat "Forest." A stop was made at Hale's Bar to permit an inspection of the work being done by the power company, as explained in the paper presented by Mr. Thos. E. Murray. The return trip to Chattanooga was by special train.



LOOKOUT MOUNTAIN

No. 1105.*

THE IMPROVEMENT OF THE TENNESSEE RIVER AND POWER INSTALLATION OF THE CHATTANOOGA AND TENNESSEE RIVER POWER COMPANY AT HALE'S BAR, TENN.

BY THOMAS E. MURRAY

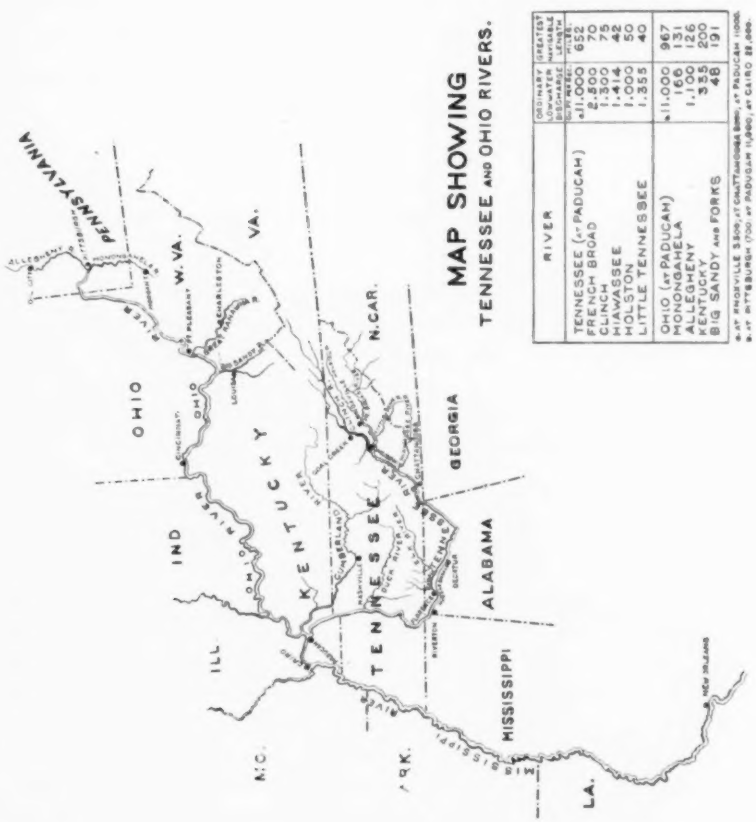
(Member of the Society.)

The Tennessee River is six hundred and fifty-two (652) miles long. It is formed by the junction, four and one-half ($4\frac{1}{2}$) miles above Knoxville and one hundred and eighty-eight (188) miles above Chattanooga, of the French Broad River which rises in the western part of North Carolina and the Holston River which rises in the southwestern part of the State of Tennessee.

Thus formed, the Tennessee River flows in a southwesterly direction across the State of Tennessee and through the City of Chattanooga. Its general course is parallel to the eastern slope of the Cumberland plateau, and it receives on the way a number of important tributaries. At Chattanooga the river inclines more to the westward and breaks through the range of the Cumberland Mountains. After passing the mountains it crosses the northern part of the State of Alabama, flows past the northeast corner of Mississippi, and turning to the north crosses the States of Tennessee and Kentucky, finally emptying into the Ohio River at Paducah, a course of 464 miles. Together with its principal tributaries, it forms a system of internal waterways capable of being navigated by steamboats more than thirteen hundred (1,300) miles. In addition to this, its tributaries are still further navigable by rafts and flatboats, for a distance of more than one thousand (1,000) miles, making a system of navigable waters of about two

* Presented at the Chattanooga meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

thousand three hundred (2,300) miles in length, with a drainage area of about forty-four thousand (44,000) square miles. The river is navigable the entire year from its mouth to Riverton, Alabama, a distance of two hundred and twenty-six (226) miles. Between Riverton and Muscle Shoals—a distance of sixty-two and one-half ($62\frac{1}{2}$) miles—the obstructions to navigation have been



RIVER	ORDINARY LOW WATER DISCHARGE (CU. FT. PER SEC.)	GREATEST NAVIGABLE LENGTH (MILES)
TENNESSEE (AT PADUCAH)	a11,000	652
FRENCH BROAD	2,500	70
CLINCH	1,300	75
HIAWASSEE	1,414	42
HOLSTON	1,000	50
LITTLE TENNESSEE	1,355	40
OHIO (AT PADUCAH)	b11,000	967
MONONGAHELA	166	131
ALLEGHENY	1,100	126
KENTUCKY	335	200
BIG SANDY AND FORKS	48	191

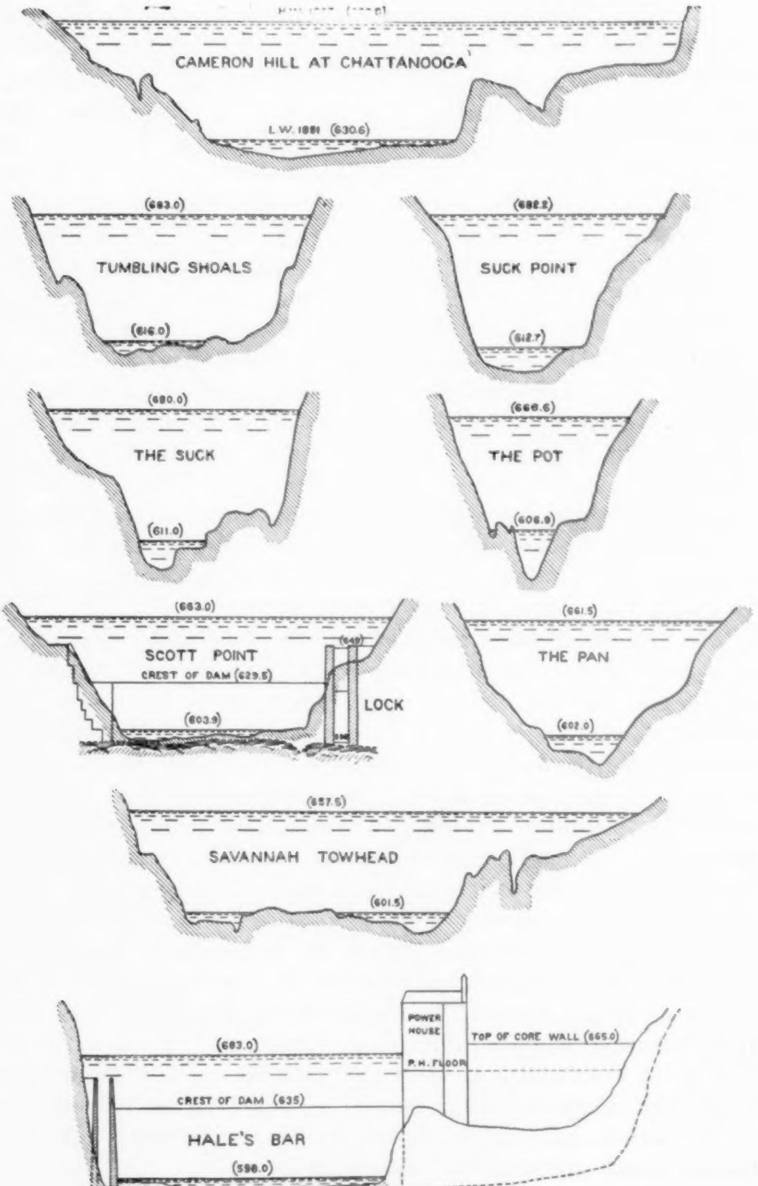
surmounted by means of canals with locks, so that a low-water channel of five (5) feet depth is available the entire year. From Muscle Shoals to Chattanooga—a distance of one hundred seventy-five and one-half ($175\frac{1}{2}$) miles—the low-water navigation is limited to a draught of water not exceeding two (2) feet, and for long periods during high-water navigation must be entirely suspended.

The chief steamboat commerce of the river consists of local boat lines having headquarters at the principal towns along the river, and there is no through traffic covering the entire system; the longest regular boat service is between Chattanooga and Paducah when the stage of water permits.

The total commerce of the Tennessee River amounted in the calendar year of 1904 to nearly 1,500,000 tons, valued at approximately thirty million ($30,000,000) dollars. Of this traffic less than ten (10) per cent. was carried over four hundred and fifty (450) miles; about fifty (50) per cent. between two hundred (200) miles and 450 miles, and about twenty (20) per cent. between fifty (50) and 200 miles. The commerce on the portion of the river above Chattanooga, in the same year, amounted to over 500,000 tons, valued at about four million five hundred thousand ($4,500,000) dollars. The commerce carried on the river between Chattanooga and Florence—a few miles above Riverton—in the same year amounted to about 170,000 tons, valued at seven million ($7,000,000) dollars; and the commerce between Florence and Paducah, in the same year, amounted to 870,000 tons, valued at over eighteen million ($18,000,000) dollars.

The general characteristics of the river are those of a broad tranquil stream with a moderate current. The bottom is usually of rock or coarse gravel, and its banks are remarkably firm and stable. The river, as a whole, presents an unusual fixity of regimen, although while passing through the mountains below Chattanooga, a stretch of perhaps 30 miles, it assumes many of the characteristics of a mountain torrent. Its course is exceedingly crooked, the slope is excessive, and, owing to the narrow and congested channel, the current is irregular and generally very rapid. With the exception of this stretch the navigation of the river presents no difficulties, and from the time of the settlement of the country it has been one of the regular highways of commerce for the region through which it flows. The navigation of this "Mountain Section," however, is quite difficult and uncertain. At low water, on account of the rapids, it shows many obstructions, while at high water it is dangerous on account of the velocity of the current and of the eddies and whirlpools caused by its irregular and contracted cross-section and its excessive flow.

Between Chattanooga and Shellmound—a distance of thirty-nine (39) miles by the river—there are ten (10) shoals at which the low-water channel depth is less than three (3) feet, and five (5) natural obstructions at which, although sufficient depth is



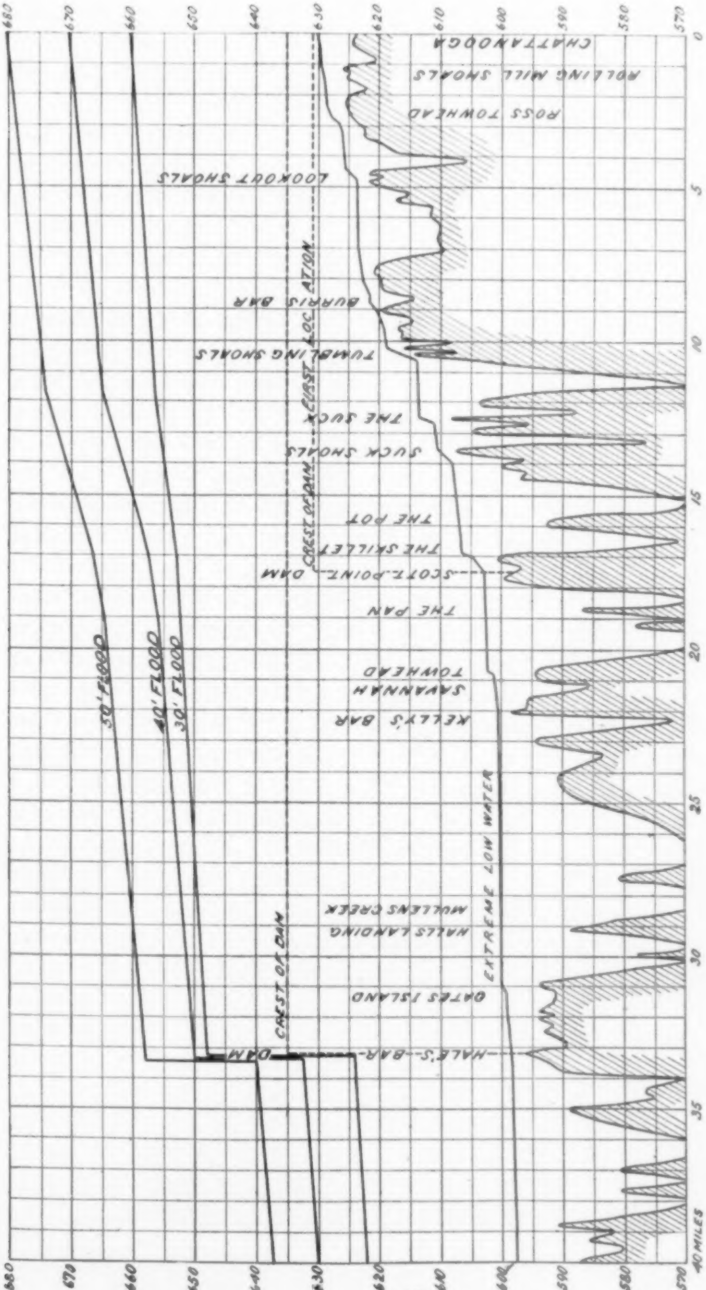
CHARACTERISTIC CROSS SECTIONS

HORIZONTAL SCALE

0 200 400 600 800 1000 FT.

0 25 50 75 100 125 FT.

VERTICAL SCALE



LONGITUDINAL SECTION OF THE TENNESSEE RIVER.

found, navigation is difficult and somewhat dangerous at nearly every stage by reason of the contracted waterway and the swiftness of the current. Maps and profiles are given showing the shape of the river and the location of the shoals.

At the foot of Williams Island—ten (10) miles below the Walnut street Bridge at Chattanooga—the river enters the Cumberland mountains, and for the succeeding eight (8) miles it is practically a mountain torrent of unusual dimensions. Completely hemmed in by the mountains, it follows a narrow, tortuous, and rocky channel feared by steamboat men and others navigating the river.

The average width of the eight (8) miles through the mountains, which may be called the "Mountain Section," does not exceed one thousand (1,000) feet at the level of ordinary high water, ranging from seven hundred (700) feet at the "Pot" to fifteen hundred (1,500) feet at "Savannah Towhead." The full significance of these figures can best be understood by recalling the fact that the ordinary low-water width of the river, where it is normal, is about equal to the high-water width above given. The great variation from the normal in the width, area and form of the cross-sections is illustrated by the cross-section plate which shows the normal sections at various places in the river. The greatest engorgement takes place in the vicinity of the "Suck," where the range between extreme high and low water is nearly seventy (70) feet. This range is reduced to sixty (60) feet at the "Pot." Between these points the fall is excessive, and the flow through the narrow and somewhat uniform channel is similar in many respects to that in a sluiceway. Below the "Pot" the river widens out and becomes practically normal at "Savannah Towhead." From the area of the sections shown and the estimated maximum discharge at Chattanooga, the mean velocity of flow has been calculated for the stages on the Chattanooga gauge, increasing by five (5) feet from zero to extreme high water. These results are given in Table 1.

THE "SUCK."

TABLE 1.

Stage Chattanooga gauge.	Discharge c. f. s. Chattanooga.	Area, in Square Feet, and Mean Velocity, Feet per Second.								Remarks.
		Chattanooga.	Tumbling Shoals.	Suck Point.	The "Suck."	The "Pot."	Scott Point.	The "Pan."	Savannah Towhead.	
0.....	8,000	1.70	5.33	2.34	3.32	2.32	4.44	2.90	3.64	Mean Velocity
		4,710	1,500	3,425	2,410	3,450	1,800	2,760	2,200	Area.
5.....	28,000	2.78	4.06	3.65	5.76	4.94	4.25	5.06	3.59	
		10,080	6,900	7,675	4,860	5,670	6,580	5,530	7,800	
10.....	52,000	3.33	4.37	4.64	6.60	6.40	4.29	6.08	3.50	
		15,600	11,900	11,200	7,885	8,120	12,110	8,560	14,850	
15.....	85,000	4.00	5.09	5.88	7.23	7.74	4.69	6.91	4.06	
		21,300	16,700	14,450	11,765	10,990	18,120	12,310	20,925	
20.....	120,000	4.45	5.59	6.70	7.59	8.70	5.19	7.15	4.42	
		27,000	21,470	17,900	15,825	13,780	23,050	16,780	27,175	
25.....	160,000	4.97	6.19	7.24	7.96	9.72	5.69	7.65	4.75	
		32,170	25,780	21,600	20,100	16,470	28,120	20,880	33,725	
30	205,000	5.16	6.64	8.27	8.38	10.80	6.14	8.30	5.20	
		39,720	30,860	24,800	24,450	19,000	33,370	24,700	39,425	
35.....	250,000	5.02	6.89	8.33	8.53	11.36	6.59	8.72	5.44	
		49,750	36,280	30,000	29,300	22,000	37,950	28,650	45,925	
40.....	310,000	5.11	7.35	9.52	9.00	12.30	6.97	9.34	5.64	
		60,700	42,170	32,550	34,450	25,200	44,480	33,200	54,925	
50.....	475,000	5.57	8.86	11.42	10.60	15.25	8.07	11.45	6.58	
		85,200	53,600	41,600	44,800	31,170	58,880	41,500	72,250	
58.....	700,000	6.76	10.95	13.95	12.92	19.18	10.30	14.42	8.18	
		106,000	64,000	50,130	54,200	36,500	68,060	48,900	85,575	

The difficulties of navigation of this "Mountain Section" were early brought to the attention of the National Government, and as far back as 1830 the first attempts at improving the channel of the river were made. The proposed improvements in this case amounted only to the obtaining of a low-water channel, having a depth of about two feet, high-water navigation of the "Mountain Section" at that time being practically impossible. This portion of the river has been examined and reported upon a number of times by the United States Engineer Officers; the first report was made by Colonel Long in 1830. The next report was by Colonel McClellan in 1853, and other reports were made in 1854, 1868, 1890, 1892 and 1898. In all of these reports it has been generally conceded that the obstructions to navigation offered by the "Moun-

tain Section" were the most serious of any to be found upon the river from Knoxville down. Colonel Long in his report outlined a plan for the improvement of the "Mountain Section," although the degree of improvement which he sought to obtain was exceedingly moderate. It appeared to have reference to a depth of two feet at low water, and it was expected that ascending boats would make use of ropes and be warped through the swift water of the "Mountain Section." The other reports submitted have generally proposed to carry the improvement a little further, to remove a greater number of boulders, to dredge a little deeper through certain bars and points, and in some cases to attempt to diminish the high-water velocity by cutting trees and removing boulders from the sides of the high-water channel. These plans have gradually been carried out from time to time, as money was available for the purpose. In all about $150,000 has been expended by the government upon this portion of the river in the construction and maintenance of the various works for channel improvement. As a result, navigation through the "Mountain Section" is somewhat less dangerous than it was originally, and it is also less difficult at the stages at which the channel can be used. However, the season of navigation has not been materially lengthened by all the work which has been done, and navigation through the "Mountain Section" is still entirely suspended at extreme low water, and the period of suspension is still very long. Navigation through the "Mountain Section" is generally considered unsafe for any boat of sufficient size to be useful for the purposes of commerce when the river falls below a 3-foot stage by the Chattanooga gauge, and the records would indicate that there is an average suspension of navigation on the river for at least three months out of every year, and this suspension occurs in the late fall and early winter, at the time when the navigation of the river would be most useful and most advantageous.

In 1891-1892 an examination was made of the reach of the "Mountain Section," under the direction of G. W. Goethals, Corps of Engineers, U. S. A., during the course of which the velocity of the current in the channel was measured at most of the points given above for six stages. The results of these observations are given in Table 2. The velocities given in the tables, while they represent in one case the mean conditions of the whole section, and in the other the actual conditions at the point of observation, do not always give an adequate idea of the difficulties of navigation at the points mentioned, on account of the complications caused by the

TABLE 2.

SHOWING OBSERVED VELOCITIES AT VARIOUS STAGES AND LOCATIONS IN THE "MOUNTAIN SECTION."

Chattanooga Gauge	5.00	7.50	8.50	9.50	16.60	32.00
Tumbling Shoals	7.50	6.15	.....	6.20	7.90	7.10
Suck Point	7.65	8.15	8.45	9.30	9.75	10.50
The "Suck"	10.90	11.05	11.65	11.45	10.00	12.20
Suck Shoals	7.40	......	......	......	.....	......
Richies Point	......	......	......	......	10.15	13.40
The "Pot"	9.30	10.50	......	11.75	13.00	15.00
The "Skillet"	10.15	9.05	......	7.10	7.00	7.25

formation of whirlpools and eddies; for instance: The table shows the highest velocities at the "Pot" and it is natural to expect to encounter there the greatest difficulties, whereas it is claimed by steamboat men that both the "Suck" and "Suck Point" are more dangerous and difficult to pass at high stags.

At low water the fall is governed by the longitudinal profile of the river bed, and is consequently concentrated at the shoals and other obstructions where, in some instances, for a limited distance, it amounts to more than one (1) foot in one hundred (100) feet. The total fall of extreme low water between Chattanooga and Shellmound is thirty-four (34) feet. This naturally divides itself into four (4) reaches with comparatively uniform fall, as follows:

Chattanooga to Tumbling Shoals	10	miles;	11.94	feet fall
Tumbling Shoals to Scott Point	7.5	"	14.8	" "
Scott Point to Kellys Ferry	5.2	"	3.6	" "
Kellys Ferry to Shellmound	16.1	"	3.8	" "
Total	38.8	miles;	34.14	feet fall

The high-water fall is largely controlled by the contracted sections in the mountains, and may be divided into three reaches over which the fall is nearly uniform, as follows:

Chattanooga to the "Suck"	12.7	miles;	7.8	feet fall
The "Suck" to Kellys Ferry	10	"	25.8	" "
Kellys Ferry to Shellmound	16.1	"	13.5	" "
Total	38.8	miles;	47.1	feet fall

Because the river is confined in a deep, narrow and crooked canyon in the mountains, and because its fall through this canyon is excessive, it was seen that the limit of improvement by channel

work had practically been reached, and in 1890 the Board of Engineers, which was appointed to consider the improvement of the "Suck," a name which is sometimes applied to the whole of the "Mountain Section," and sometimes is limited to only one of the obstructions, reported that the only complete and practical improvement of this section of the Tennessee River would be by the construction of canals, or by arrangements for slack-water navigation, but they report further that the great expense of slack-water navigation rendered it unworthy of consideration at that time. Since that time the project of slack-water navigation of the "Mountain Section" has been repeatedly taken up, and a number of proposals have been made by the government. In 1900 the government engineers reported on a system of slack-water navigation, which they estimated would cost in the neighborhood of one million dollars.

In planning a system of slack-water navigation for the "Mountain Section" a very serious difficulty is met with at the outset, and that is the enormous flood height which the river occasionally, although at rare intervals, attains in this particular place. As before stated, the banks of the river rise rapidly from the low-water channel. There is no flood plain, so that even at the highest stages the surface width of the river in some places in the "Mountain Section" is not more than 1,000 feet, although this does not exceed its average low-water width at and above Chattanooga. The consequence is that in time of flood there is an engorgement of the waters at this narrow point, and the water is backed up and held as by a dam until it has been known to attain a height in the mountains of 70 feet above its ordinary low-water level. This engorgement ponds the water, and diminishes the high-water slope for many miles above Chattanooga. It is true that such exceptional flood heights are of very rare occurrence, only one authentic record of such a flood being in existence.

At such extreme floods down along the river the banks are generally inundated, bottom lands are all overflowed, the landings are under water, and it is a matter of indifference whether navigation is possible or not. It would seem, therefore, unwise and unnecessary to attempt to provide safe and easy navigation for such extreme and exceptional floods. The most difficult problem presented for solution in connection with the installation of a system of slack-water navigation has been the determination of the "guard" for the locks; that is, the height that the lock must have above the dam in order that it may continue in use until the dam

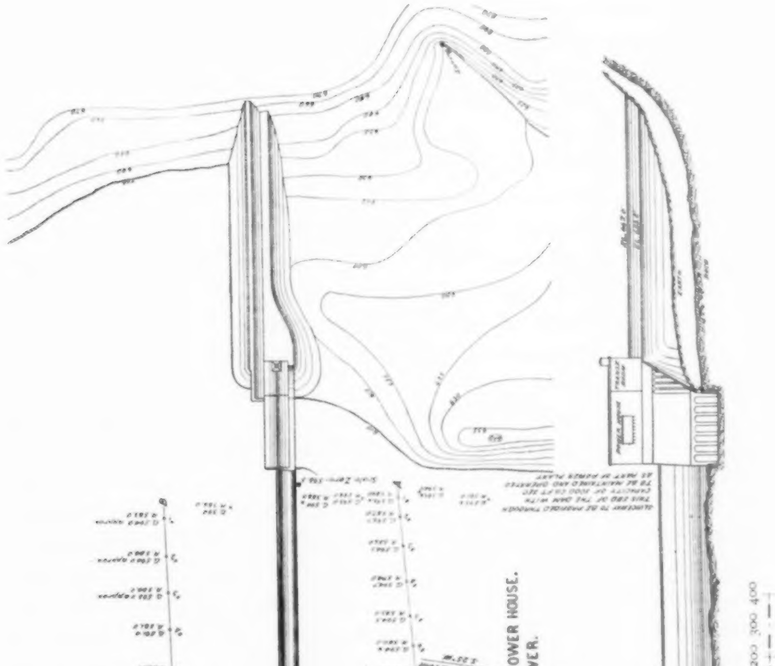
is so completely submerged that vessels may safely and easily pass over it.

Owing to the very narrow water way in the "Mountain Section," and to the great velocity of water at high stages, it seems doubtful if any dam of useful height would be submerged at any stage of the river, and it would therefore be necessary, in order to insure an entirely uninterrupted navigation at every conceivable stage of the river, to construct locks which could be operated at all stages, and this would involve a height of walls and of gates that would render the cost excessive. The alternative is to admit of a possible suspension of navigation during the time of great floods, knowing these periods of suspension must be short and of rare occurrence.

An examination of the hydrographs from 1875 to 1900 shows that if the stage of 35 feet on the Chattanooga gauge is assumed as the limiting height beyond which it will not pay to attempt to provide navigation, then in the past 25 years there would have been 17 suspensions of navigation, amounting in the aggregate to 85 days or a little more than 3 days per annum.

Records show that there were actually 2,317 days during which navigation even for very light draught boats was suspended, and probably quite as many more days when it was attended with difficulties and dangers that a slack-water improvement would have entirely obviated. During this period of 25 years there were 12 years in which the height of 35 feet on the Chattanooga gauge was not reached at all.

Thirty-five feet on the Chattanooga gauge was, therefore, assumed as the height up to which it must be possible to operate the locks. Having fixed upon this height, a number of different plans were considered: A plan to improve only the worst part of the "Mountain Section" by a dam which would back the water up over Tumbling Shoals; a plan to improve the entire reach from the "Skillet" to Chattanooga by a higher dam; a plan considering a site near the Savannah Towhead for a lock and dam to accomplish either of the above purposes; the plan finally reported being a single lock and dam in the vicinity of the "Skillet," the dam to have such a height that it would back the water up to Chattanooga and to secure at the lowest stages a navigable channel not less than 5 feet in depth for the entire distance, and to use with this dam a single lock, the walls and gates of such height that it could be used until the river reached a stage of 35 feet on the Chattanooga gauge.


GENERAL PLAN OF LOCK, DAM, AND POWER HOUSE.
HALE'S BAR, TENNESSEE RIVER.
SCALES
HORIZONTAL 1" = 800'
VERTICAL 1" = 40'
Horizontal Scale
100 200 300 400

As might be expected in a place where the river has cut its way down 1,000 feet through rock of varying hardness, the bottom of the river is not composed, as a rule, of solid rock, but is made up of boulders, gravel and drift, so that considerable difficulty was experienced in finding a suitable foundation for a lock and dam. The locality known as "Scott Point" was finally selected, and after several hundred borings it was finally demonstrated that a suitable rock foundation at a reasonable depth could be had both for the lock and the dam.

The work proposed in the report of 1900 consisted of a lock of cut-stone masonry, 65 feet wide in the clear and 300 feet long between hollow quoins. The dam was to be constructed of heavy timber cribs filled with stone, the crest of the dam to be perpendicular with the general direction of the current, and to be horizontal and straight, the deck to slope downward each way from the crest at a slope of two to one.

About this time, some of the business men in Chattanooga, who had been following the progress of the development of water power in various parts of the country, conceived the idea that these works for the amelioration of the traffic conditions on the Tennessee River might be made to pay for themselves, by the conversion of the water power generated at the dam into electrical energy; and the whole matter was taken up with a great deal of earnestness, notably by C. E. James and J. C. Guild. The scheme was examined in all its bearings, especially as to its influence on the development of the industrial situation in Chattanooga, and the aid of the Honorable John A. Moon, congressman from that district, was enlisted to obtain the necessary legislation.

These efforts culminated in an Act of Congress approved April 26, 1904, which authorized the Secretary of War to grant permission to the City of Chattanooga to build and construct a lock and dam across the Tennessee River at Scott Point, near Chattanooga, Tennessee, under his direction and control, in accordance with plans and designs made by Major D. C. Kingman, Corps of Engineers, United States Army. This act also provided that if the City of Chattanooga should fail within four (4) months from the date of the passage of the act to notify the Secretary of War of its intention to construct the lock and dam, then the Secretary of War was empowered to offer the franchise to C. E. James and J. C. Guild, residents of Chattanooga, Tennessee, for a further period of eight (8) months, and failing to contract with them, to contract with any private corporation, company, firm, or business,

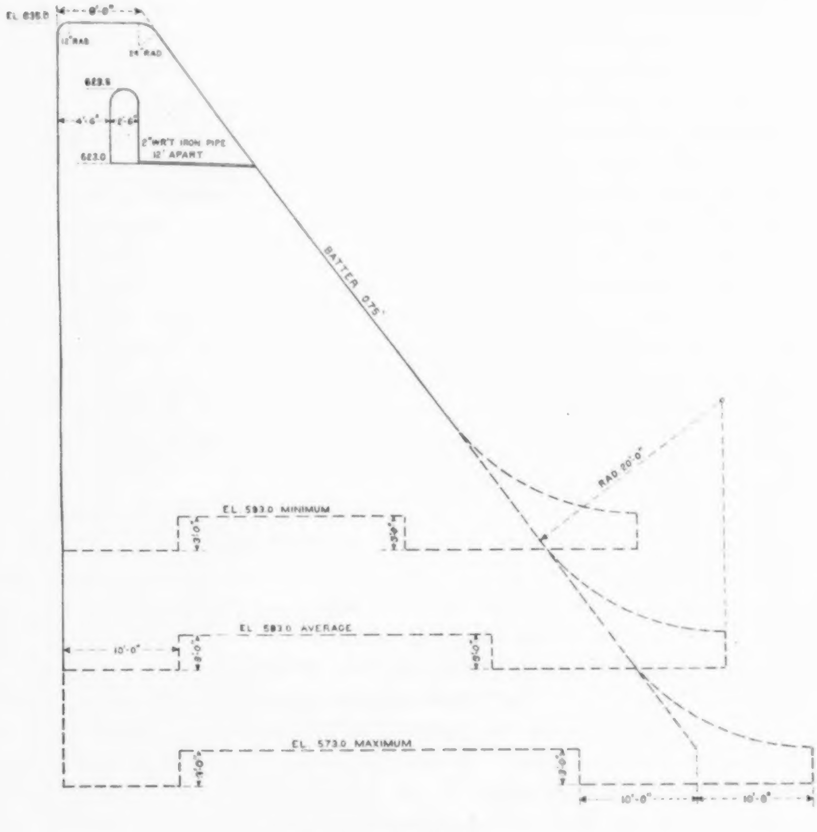
for the construction of the lock and dam on the terms and in the manner provided. The parties obtaining the franchise were to construct the lock and dam at their own expense, acquiring the land that might be necessary for that purpose. The United States was to furnish the machinery for the lock, but the power for the operation of the same was to be furnished by the company. And it was further provided that the utilization of power by means of the dam was not to interfere with the flow of water or the navigation of the river.

The City of Chattanooga failed to take advantage of the opportunity; but Messrs. James and Guild, seeing the advantages that would accrue to themselves and their associates, organized the Chattanooga and Tennessee River Power Company to undertake this work, with Mr. A. N. Brady as the leading spirit of the financial group. The active interest of Mr. A. N. Brady in industrial enterprises of this character and his appreciation of their possibilities has made feasible the development of many similar undertakings; he had associated with him as his technical advisers in this work Mr. John Bogart and the author of this paper. The Chattanooga and Tennessee River Power Company then entered into contract with the United States Government for the construction and maintenance of the works.

The question to be considered by Major Kingman when deciding upon a location for a lock and dam was one of economy; and a location which while giving ample navigation facilities would require the smallest expenditure of funds, was, therefore, the one to be sought, and it was found at Scott Point. When, however, Congress passed an act allowing private parties to build the lock and dam, in return for the use of the water power for ninety-nine years, a different aspect was put upon the case, and the paramount question was not economy. Provided that the interests of navigation were fully safeguarded, it was desirable to locate the works lower down the river and get the benefits of the extra fall in such distance. The use for ninety-nine years of the extra power so gained would far more than compensate for the extra cost of the structures, due to increased height.

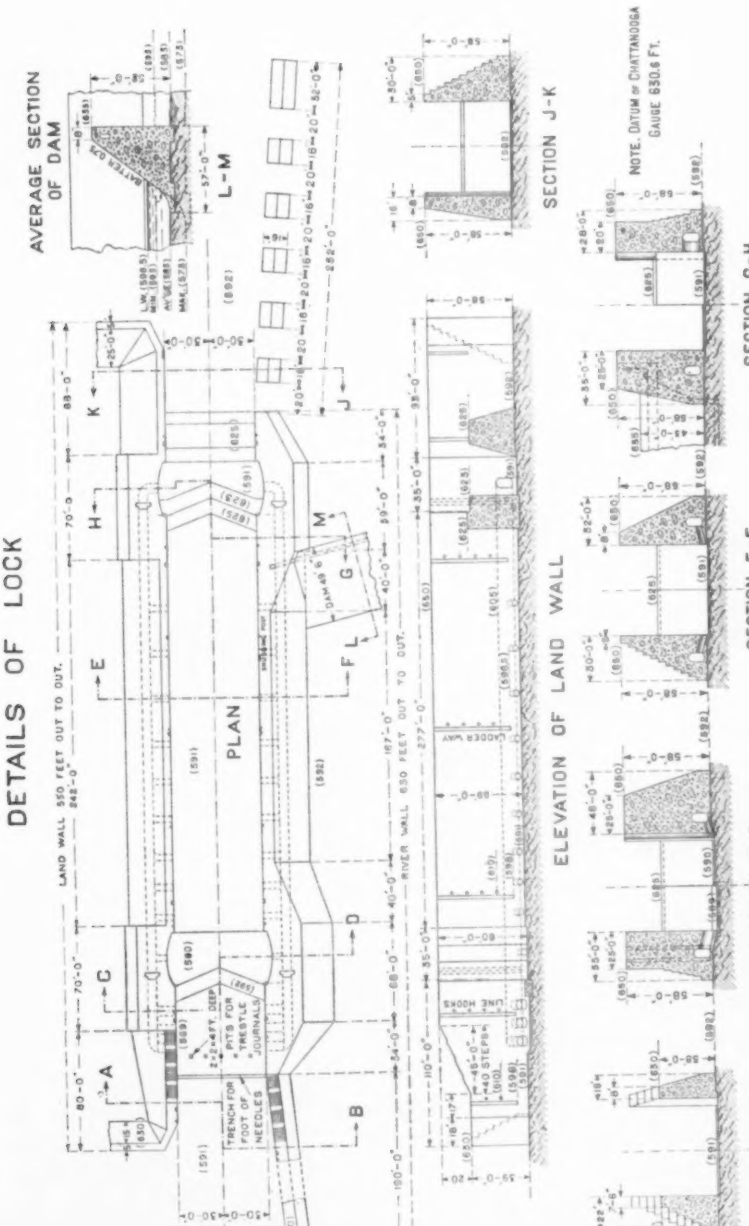
The original act of Congress in relation to this construction fixed the location of the dam and lock at Scott Point, about 16 miles, along the course of the Tennessee River, below the City of Chattanooga. A study of the conditions affecting the river, particularly in the higher stages of flow, showed that the head upon the turbines would be greatly decreased as the river rose. This would

be caused by the fact that below Scott Point there occur several narrow passes which in flood stages set the water back to such an extent that with a dam giving a head of over 35 feet at low stages there would be, at a flood of 25 feet, only about 17 feet head on the wheels. Also that at the time of a flood of 35 feet, which



CROSS SECTION OF DAM

occasionally occurs, the head would be reduced to about 12 feet, and that at a stage of 40 feet, which may occur, the head would be only 10 feet. The difficulties of securing a satisfactory development under these conditions were so great that after representation of the facts to the authorities at Washington an act was passed by Congress (approved January 7, 1905) authorizing the location of the dam at such other point or place in the mountain section


DETAILS OF LOCK
PLAN
AVERAGE SECTION OF DAM
L-M
LAND WALL 550 FEET OUT TO OUT.
RIVER WALL 630 FEET OUT TO OUT.
TRENCH FOR FOOT OF NEEDLES
PITS FOR TRESTLE JOURNALS
ELEVATION OF LAND WALL
LADDER WAY
LINE HOOKS
40 STEPS
SECTION J-K
SECTION A-B
SECTION C-D
SECTION E-F
SECTION G-H
NOTE. DATUM OF CHATTANOOGA GAUGE 630.6 FT.

of the river below Scott Point as the Secretary of War might approve.

A location was then studied at Kellys Bar, 5½ miles below Scott Point. This was found to be a favorable place for the construction of the requisite works, but there being still a number of narrow places in the river below, the hydraulic conditions, while better than at Scott Point, were not satisfactory.

A location was finally found at Hale's Bar, 33 miles below Chattanooga, which is satisfactory from all points of view. The foundations for all structures are on solid rock, the river at this point leaves the mountain gorge and enters upon a wide valley, so that the backing up of flood waves is much less than at points above. The head on the turbines at low water will be 39½ feet. At a flood of 25 feet there will be 27 feet head, and at a flood of 35 feet there will be 21½ feet head, and at a stage of 40 feet the head would still be about 19 feet, thus assuring a large continuous output of power under all conditions of flow.

It has also been found practicable, with the sanction of the Government, to make the crest of the dam higher at this point than at the other places studied. The increased pool, extending to Chattanooga, gives very desirable and satisfactory storage for use in regulating power at times of low flow and also affords good navigation at several bad points not improved by constructions at the other locations. The Scott Point lock was to be of cut stone masonry, and the dam of timber cribs filled with stone. The Hale's Bar lock is designed to be of concrete; the dam also is to be of concrete, as being more durable and water-tight. The natural fall in the river at low water between Scott Point and Hale's Bar is five feet, and the crest of the dam, as now designed, is six and one-half feet higher than the Scott Point design, giving an extra head of eleven and one-half feet at extreme low water. With a discharge of 5,000 cubic feet, this means that the present plan will deliver at Chattanooga about 5,000 horse-power more than the old plan and location.

The lock and dam were designed under the direction of Major H. C. Newcomer, Corps of Engineers, U. S. A., by John M. G. Watt, Principal Assistant Engineer. The concern of the Government being only the conservation or improvement of the navigability of the river, it required that only the lock and dam be designed by the engineer officer in charge of the Tennessee River; the power house and all appurtenances for developing the water

power were to be designed by the grantees, subject, however, to the approval of the Secretary of War.

The designs for the power plant have been developed by John Bogart, C. E., who had many novel features to encounter and difficulties to overcome, the chief being how to deliver not less than a certain fixed minimum of power every hour in the year, irrespective of the stage of the river. In this connection I cannot do better than quote the words of Mr. Bogart: "The difficult hydraulic problem in the development of this power arises from the variations in volume of flow and in head upon the turbine wheels. The variation in volume runs from about 5,000 cubic feet per second up to 255,000 cubic feet per second, with the probability of an occasional flood reaching a volume of 320,000 cubic feet per second. Once during the past twenty years there was an unprecedented flood with a volume of possibly 600,000 cubic feet per second; the duration of this was brief.

"A rather general statement is that for an approximate average period of about two months of a year the flow will be between 8,000 and 16,000 cubic feet per second; for about four months between 12,000 and 59,000; for about four months between 16,500 and 59,000; for about two months between 21,500 and 92,000 cubic feet per second. The successful operation of the generators which transform the water power into electrical current requires that the speed shall be substantially constant at all times. If the variations in volume and head were not greater than those found it would not be difficult to secure this uniformity of speed. But the volume of flow has in the past and probably will in the future be at times less than 8,000 cubic feet per second, and this low flow may continue for a number of consecutive days, possibly for several consecutive weeks. To insure commercial success in the enterprise it is necessary that the electrical output should continue during this period. Therefore, the rate of speed of the turbine must continue uniform, and as the volume is limited the turbines must be designed to secure the highest efficiency during these periods of low flow, and it is also important that the head upon the wheels should then be as great as is in any reasonable way practicable.

"It is also the fact that there have been and doubtless will be periods when the volume of flow is considerably greater than the 92,000 cubic feet per second, which is the ordinary high-water flow. The flow has frequently exceeded that volume for many consecutive days; 157,000 for a week; 189,000 for a similar

period; and the volume has reached 320,000 more than once. Each of these additions of volume involves a reduction of head which will be lowered to less than 18 feet when these larger flows occur. To secure a uniformity of speed and a regular output of power under these conditions I have found it necessary to design a third turbine wheel upon the shaft on which the two turbines will be fixed for the lower volumes. This third wheel will be adapted to the utilization of great volume at the lower heads, there being then ample water which can better be utilized through the wheels than allowed to pass over the dam."

The lock and dam will be built of cyclopean concrete; that is to say, large stones, say up to ten tons weight, or larger, if the machinery can handle it, will be embedded in and completely covered with concrete to a depth of not less than nine inches; so that the body of the structure is chiefly of uncut blocks of stone, laid in random range, and separated from each other in every direction by nine inches or more of concrete. This will cheapen the construction, and acually be safer than simple concrete, as it will make a heavier mass, while with ordinary care in laying it will be just as impermeable.

The lock is located on the west or right bank of the river. It will be built against a rock bluff, thus obviating any danger from the river cutting around during high water. The dam will be 1,200 feet long and extend from the lock to the power house. The power house will be about 200 feet long and will be built as a continuation of the dam. The power house will be connected with the left bank of the river by means of an earth embankment with a concrete core wall. This core wall will extend to solid rock, will have a width of 4 feet on top and a maximum width of 8 feet at the bottom. Its top will be at elevation 665. The earth bank will be carried two feet higher, will have a top width of 12 feet, and will have side slopes of two to one. At elevation 653 it will have a berm on the lower side which will carry an approach to the power house. The total length between the rock bluff and the hill where the embankment will terminate is about 2,300 feet, of which the dam comprises 1,200 feet, the power and transformer houses about 300 feet, and the embankment about 700 feet.

The tops of the lock walls will be at elevation 650. The inner or land wall proper of the lock will be 427 feet long, with approach walls aggregating 123 feet more, or a total length over all of 550 feet. At each end there will be a wing wall running into the bank at the same elevation as the top of the main wall. The length

of the river wall proper is 440 feet, with a lower approach wall 190 feet long. The upper approach wall is formed of detached piers 16 feet long, with spaces of 20 feet between them and extending 232 feet upstream of the upper end of the lock. The height of the lock walls will be about 58 feet. The land wall will have a bottom width of 30 feet and a top width of 5 feet. The bottom and top widths of the river wall will be 32 and 8 feet respectively. At the buttresses supporting the gates the top width is increased in all cases to 25 feet, the base of the land wall to 33 feet, the base of the river wall to 35 feet at the upper end, and to 46 feet at the lower end.

The lock will have a clear width of 60 feet. The gates will be of the mitering type, horizontally framed, of mild steel, and will be opened and closed by rack bars operated by electricity, with arrangements for hand power in case of failure of the electric current. These gates will be remarkable for the head they will have to support. In extreme low water seasons, with flashboards on the dam, the difference in elevation between the two pools will be about 40 feet. Each leaf of the lower gate will be about 34 feet long by 59 feet high, and will weigh about 129 tons; each leaf of the upper gate will be about 26 feet high, and will weigh about 50 tons.

The lock chamber will be filled by two culverts, about 11 feet by 6 feet, one in each wall, running the whole length of the chamber, and having ten openings three feet below the level of low water. It will be emptied by means of two culverts of the same size, each having three openings into the lower bay. These culverts will be operated by Stoney sluice gates, operated from the top of the wall by electric or hand power. The chamber will admit at low water a fleet of boats or barges drawing six feet, with a width of 59 feet and a length of 300 feet.

In case of accident to the gates the lock can be closed by placing five steel trestles across each end, fitting into journals placed during construction. These trestles would then be connected by steel beams against which would rest needles or vertical beams of timber or steel.

The crest of the dam will be at elevation 635 and have a width of eight feet. The upstream face will be vertical while the downstream face will have a batter of three horizontal to four vertical, terminating in a curve with a radius of 20 feet. The height will vary from 42 feet to 62 feet, depending on the elevation of solid rock, with an average height of about 52 feet. Running the full

length of the dam there will be a passageway, as shown in the accompanying section. This will be two and a half feet wide by six and a half feet high. It will terminate in the land wall of the lock and in the power house in shafts extending above extreme high water. At the bottom of this passageway at intervals of 12 feet will be placed two-inch wrought-iron pipes extending out to the downstream face of the dam. These pipes will supply air under the falling sheet of water passing over the dam, and prevent the formation of a vacuum. This passageway will also be used as a means of crossing from the power house to the lock, and will carry the wires for furnishing the electricity for operating the machinery of the lock and for lighting the lock and other United States property. Near the power house the dam will contain a sluiceway to supply water to the lower pool at times when there is none passing over the dam or through the power house.

The power house will consist of seven bays, each containing two units. Each unit will consist of three turbines on a vertical shaft, carrying the generator at its upper end. Under ordinary stages of the river only two of the turbines will be used, the third being held in reserve and used when there is a large quantity of water flowing, but giving a reduced head. Each generator will have a normal capacity of 2,250 to 3,000 k.w. The main floor of the power house will be at elevation 653. Ten feet below this will be a floor carrying the supports of the rotating parts of the turbine and generators and also the governors.

The water will be conducted to the power house by a headrace excavated in the bank of the river. The tailrace will also be excavated in the bank of the river, and will extend down into the rock, its elevation at the power house being 571. Low water below the dam will be at elevation 598.5. This will give a head of 36.5 feet, which can be increased three feet by the use of flashboards.

The lock, dam and power house are all to be built of concrete.

Although nothing has been definitely decided as yet regarding the electrical apparatus, it is the intention at the present time to generate current at 6,600 volts, 60 cycles, and step up through oil-insulated, water-cooled transformers to either 23,000 or 40,000 volts for the transmission line. The first transmission line will probably consist of two three-phase lines on the same pole, and will be carried in a straight line over the mountain to the south of the bend in the Tennessee River just below Kellys Ferry.

From this point the line will follow the carriage road through the hills to Chattanooga. The Receiving Sub-Station will probably be equipped with 23,000 or 40,000 volt air-blast step-down transformers, with 2,300 volts distribution within the city.

It is expected that the entire plant will be completed and ready for operation about October 1, 1907. In conclusion, credit should be given for data and other help in the preparation of this paper to John Bogart, Consulting Engineer, and J. C. Guild, Chief Engineer of the Chattanooga and Tennessee River Power Company, and to C. E. James, Major H. C. Newcomer, Engineer Corps, U. S. A., and John M. G. Watt, Principal Assistant Engineer; to the reports of the War Department, and to my assistant, Geo. A. Orrok, Member A. S. M. E.

DISCUSSION.

Mr. Donnelly.—I wish to make a few remarks on the general proposition and to call attention to the harmony between the general government and the engineering development of water ways.

We are undoubtedly going to see much more of this in the future, and it seems to me to open up a field which is very desirable. The utilization of the interior water ways of this country is almost undeveloped. Before the advent of the railroads it had an extreme development, and the original transportation development of the country was entirely a matter of the development of the water ways. With the advent of the railroad, however, the change was so great that, to a large extent, the utilization of the water ways was superseded. It is not an economic truth that the railways can entirely supersede the water ways; and as the forced development of the country is nearly finished, we are now coming to a more complete development of the country, to accomplish which it will be necessary to utilize all possible means of communication. This is a particular instance where the endeavors of the Government to promote the use of the water ways for commerce has resulted in a very much greater utilization for power purposes. In all probability the attention of the engineers to the power and the commercial end of this proposition would not have been secured if the Government had not made elaborate plans and surveys and gathered a large amount of data which has shown the possibilities in connection with this river. We understand that the control of the Mississippi River, or of any of

these large Western rivers, means an exceedingly large expenditure of money and energy in controlling the head waters, and the amount of information gathered here and being applied to other streams and distributed among engineers will tend to the working out of other situations where both navigation purposes and power purposes can go hand in hand. So far as I know, this is the first instance in which they have worked together. The building of Government dams for irrigation purposes in the West is rather a different problem, but in connection with those there will be manufacturing growth as well as agricultural growth. Modern engineering tends to introduce many new factors, and the more factors properly introduced the more possibilities there are in working out the problem both for a commercial success and engineering success. I think this should be made very clear in respect of this particular undertaking.

Mr. H. M. Lane.—As I understand it, the Tennessee River is more nearly in its primitive condition than most of our Northern rivers; that is, the timber on the mountains at the head waters has not been cut to any large extent. I should like to know if any calculation has been made with regard to the general run off that will occur after the timber has been cut. It has been found in the case of most Northern rivers that the cutting of the timber has resulted in disastrous floods at certain seasons and a very much reduced run off the balance of the year.

Mr. J. C. Guild.—I would like to state the conditions as they exist in regard to the forestry of the head waters of the Tennessee River. I think they are rather peculiar. The drainage area of the Tennessee River is very mountainous and is heavily timbered, the trees being of many varieties. This timber has been cut for the last twenty-five years, and still there is very little change so far as the effect on the flow of the river is concerned. Of course, many of the larger trees have been removed from these forests, but the young growth which has followed the removal of the larger timber has practically the same effect so far as the flow of the river is concerned. The lumberman goes into this forest for timber of a certain variety, and removing this alone, leaves the other variety still standing; later, as another variety is in demand, he goes again to cut this, but in the meantime the young trees have grown, so by this method of removal of the timber the mountains are never bare of forest. In this respect the forestry of the basin of the Tennessee River is quite different

from the White Pine forests of the North or the Yellow Pine forests of the South. In both of these cases the lumberman removes the forest entirely at one cutting.

We have taken all this into account and we do not anticipate any serious change in the flood periods or low-water periods of the Tennessee River.

President Taylor.—I would like to call attention to one previous case of co-operation between the Government and private enterprise in the development of a river which, however, has resulted variously through a term of years, sometimes with most satisfactory results and at others the reverse. And that is on the Fox River in Wisconsin. For a number of years the Fox River has practically been, so far as navigation is concerned, of very little use; the traffic on it has been but small. Its effect on freight rates, which after all is not the most unimportant feature to be considered, has been practically nothing. However, in the case of the Fox River, the unsatisfactory conditions have been at times entirely due to the particular administration on the part of the Government officers stationed in the Fox River Valley, because on their part there has been at intervals lack of co-operation and appreciation of relative importance to the community of navigability of the river and the water power. Now I am quite sure from what everyone here says that nothing of that kind will ever happen here. But at various times in the Fox River Valley it has been a very sore point between the large manufacturing interests there, with the great number of paper and pulp mills depending almost entirely for their power on the river, and the Government agents, some of whom have helped the manufacturers in every way, and others of whom have held to the strictest interpretation of the law; and for the sake of a few pleasure yachts, to enable them to get access to the lakes above, and a most insignificant traffic on the river, have held up all manufacturing along the river. As I say, I do not believe anything of that kind can ever happen here, because the yachting element at Chattanooga will probably never be very large.

Major Newcomer.—It may possibly be of interest to the members of the Society to know of another instance of co-operation that is on record. In the improvement of the Cumberland River the Government had a project for the canalization of the river from its mouth up to Rock Castle River in Kentucky, about thirty miles above Burnside, the point where the Cincinnati Southern

Railroad crosses the Cumberland River. Up to Burnside there are four or five months of the year when boats can run, but low-water navigation could not be attained except by canalization. Then above Burnside there are very bad shoals which could not be passed with safety at any time. There was a project for the canalization of that portion of the river because of the fact that it would tap a very rich coal field. The Government project, however, was not sufficiently rapid in its development to satisfy the private parties interested in the coal mines, and at the last session of Congress these parties appeared before the Committees and secured an appropriation for one lock and dam just below Burnside, which would give a pool below these bad shoals, and then agreed to go ahead and improve the shoals themselves by putting in locks and dams, and a charter was granted for that purpose. They were to build these locks and dams, on plans approved by the Government, and were to have the use of the power, of course, developed by the dams, and were allowed to charge tolls for a certain number of years. The Government has gone ahead and is now building the lock and dam below Burnside, but nothing has been actually done as yet with the development of the power scheme. A plan was prepared there for a dam 90 feet high. The discharge there is not very accurately known. The low-water discharge is something like 600 or 700 feet per second, I think. The company, which is called the Cumberland River Improvement Company, is authorized to proceed with this work; but they are not required to begin until 18 months after the Burnside lock and dam are completed. There is some talk of the development of the power for the electrification of the Queen and Crescent Railroad. I believe on the Muskingum River, in Ohio, there is also water power that has been developed by Government dams and leased to private parties. There is a lock on the Cumberland River near Nashville where Congress authorized leasing the power to private parties, but no one appears to want it. In fact, in most cases of this kind, a power plant would be exposed to frequent interruptions, since the fall over the dam is practically eliminated at all higher stages of the river.

Mr. Geo. B. Stetson.—Coming from the North, where our real estate is not in as rapid motion as it is down here, I would like to know if there has ever entered into the thought of this proposition whether this dam will not be the building of an enormous settling basin for the State of Tennessee where its real estate will be

stopped on the road to the Gulf, and whether we shall not be transferring one section of the State to another.

Mr. Hunt.—I believe Napoleon had a problem of this kind when he was circulating through Europe. He claimed Holland because it was the sediment that had been washed down by the Rhine, and he owned the Rhine.

Mr. Stetson.—Then I suppose New Orleans would claim Chattanooga for the same reason.

President Taylor.—No; Chattanooga, if that rule holds out, would own New Orleans.

Major Newcomer.—I think there is no question but there will be considerable deposit formed above this dam. That I believe has been encountered in all cases where locks and dams have been put in. At the same time I understand that the general result has not been serious—that is, that the deposit grows only to a moderate degree. The time of the maximum transportation of material carried in suspension that way, is, of course, during high flood time, and during that time the dam is going to have a relatively small effect. There will be times probably when we will have to dredge the channel above the dam. I do not believe, however, that we are going to obtain any very large share of the real estate of the counties above the dam.

Mr. Orrok.—There are one or two things that I would like to say in regard to the Tennessee River and the dam and lock proposition. When this scheme was first brought to my attention I, with most Northern people, supposed that the Tennessee River was a little ordinary Southern river, such as we have seen around Atlanta, that carries lots of sediment, that a man can wade over if he wishes to cross, and which runs pretty dry in the summer time; but in looking into the matter I have come to something like this: I find that the Tennessee River at Chattanooga has an average flow of 41,500 cubic feet per second. The lowest flow reported, which is the lowest in eighty years, is about 4,300 cubic feet per second. The maximum flow that has been reported is about 500,000 cubic feet per second. I think there is a flow of about 700,000 estimated, but that we do not know for sure. Now compare that with some other rivers. Take the Merrimac River in New England. At Lawrence that river has a low-water flow of 1,800 feet per second, and an average flow of 6,700 feet per second. That is about one-eighth of the Tennessee River flow at Chattanooga. The Connecticut River at Hartford

has a low-water flow as large as the Tennessee, but the average flow is only 17,000 cubic feet per second, and the maximum flow is less than half of the Tennessee. The Mississippi River at St. Paul has an average flow of about 18,000 cubic feet. The Susquehanna River, below Harrisburg, has a mean flow of practically 38,000 feet per second, but its low-water flow is less than half of the Tennessee. Now, as a comparison with some other larger rivers, I could not find anything in the United States to compare it with. As far as I can ascertain, there are no bigger rivers than the Tennessee. Of course, the Mississippi and Missouri are bigger. The Ohio is smaller. The low-water discharge at Paducah is about the same as the Tennessee, but the high-water discharge is a little bigger for the Tennessee than it is for the Ohio at that point.

Major Newcomer.—The low-water discharge at Pittsburgh is only about one-third of the low-water discharge of the Tennessee at Chattanooga. It is 1,700 at low water at Pittsburgh.

Mr. Orrok.—Comparing the Tennessee with foreign rivers that are bigger, the only one that I can get any accurate figures on is the Nile in Egypt. The average flow of the Nile at Assouan is 107,000 cubic feet per second. The minimum flow is practically 15,000 cubic feet per second, and the maximum 427,000. This is the largest flow on the Nile. The river Indus in India has an average flow of about 185,000 cubic feet per second and a minimum flow of 30,000 cubic feet per second.

I think this is the first case of the development of power on a navigable river in this country; that is, I do not recall any other instance where a navigable river has had a dam put in it primarily for the development of power.

President Taylor.—How about the Fox River in Wisconsin?

Mr. Orrok.—That is not a navigable river, as I understand it, properly speaking. I have a list here of power plants on rivers. The Chattahoochie River at West Point, Georgia, has a low-water flow of 830 feet per second and an installation of 13,000 horse-power. The Merrimac River, at Lawrence, with a low-water flow of 1,800 cu. secs—I haven't the horse-power here. The Mississippi River, at Minneapolis, 2,000 feet per second; I lack the horse-power there. On the Susquehanna River, with a low-water flow of 2,600, 40,000 horse-power could be installed. That is problematical yet, as they have not decided on the plant. The Connecticut River, at Holyoke, with 3,460 cubic feet per second low-

TABLE 3.

River and Location.	Drainage Area.	Maximum Flow.	Average Flow.	Minim'm Flow.
	Sq. Miles.	Cu. secs.	Cu. secs.	Cu. secs.
Little Tennessee River, Judson, N. C.	675	38,000	1,500	280
French Broad River, Asheville, N. C.	987	7,800	2,500	650
Merrimac River, Lawrence, Mass.	4,553	18,900	6,750	1,800
Connecticut River, Holyoke, Mass.	8,660	42,300	13,850	3,460
" " Hartford, Conn.	10,234	208,000	17,570	5,200
Mississippi River, St. Paul, Minn.	36,085	72,000	18,000	2,000
Susquehanna River, Harrisburg, Pa.	24,030	543,000	38,350	2,600
Tennessee River,Chattanooga, Tenn.	21,418	446,000	41,500	4,300
" " Paducah, Ky.	44,000	....	70,000	11,000
St. Mary's River, Soo St. Marie, Mich.	76,100	160,000	77,000	45,000
Nile River, Assouan, Egypt.	1,161,000	427,000	107,360	14,833
Indus River, India.	200,000	800,000	185,000	50,000
St. Clair River, Michigan.	213,900	270,000	198,000	110,000
Niagara River, New York.	254,700	260,000	220,000	175,000
St. Lawrence River, New York.	287,700	330,000	252,000	18,500
Ohio River, above Cairo, Ill.	....	845,000	....	22,000
Missouri River, St. Charles, Missouri	....	450,000	100,000	30,000
Mississippi River, above St. Louis.	....	350,000	....	100,000
" " Columbus, Ky.	....	1,600,000	440,000	160,000
" " at New Orleans, La.	1,214,000	2,000,000	....	255,000

TABLE 4.—WATER POWER DEVELOPMENTS.

	Maximum Flow.	Mean Flow.	Minimum Flow.	Installat'n H. P.
	Cu. secs.	Cu. secs.	Cu. secs.	
Chattahoochee, West Point, Ga.	57,400	....	830	14,000
Merrimac, Lawrence, Mass.	18,900	6,750	1,800	....
Mississippi, Minneapolis, Minn.	72,000	18,000	2,000	....
Susquehanna, McCalls Ferry Co.	543,000	38,350	2,600	40,000
Connecticut, Holyoke, Mass.	42,300	13,850	3,460	24,000
Tennessee, Chattanooga, Tenn.	446,000	41,500	4,300	54,000
Rhone, Lyons, France.	215,000	....	6,000	22,000

water flow, has 24,000 horse-power. The Tennessee River at Chattanooga with 4,800 low-water flow, 54,000 horse-power could be installed. Then I have one foreign river, the Rhone, near Lyons, France, with a low-water flow of 6,000 feet per second, 22,000 horse-power installed. These are the horse-powers installed. I have no figures for the mean or maximum horse-power obtained. What I was after was a little comparison that would show the size of the Tennessee River in connection with these other plants.

Major Newcomer.—With reference to the discharge, the maximum discharge was figured and estimated a few years ago as

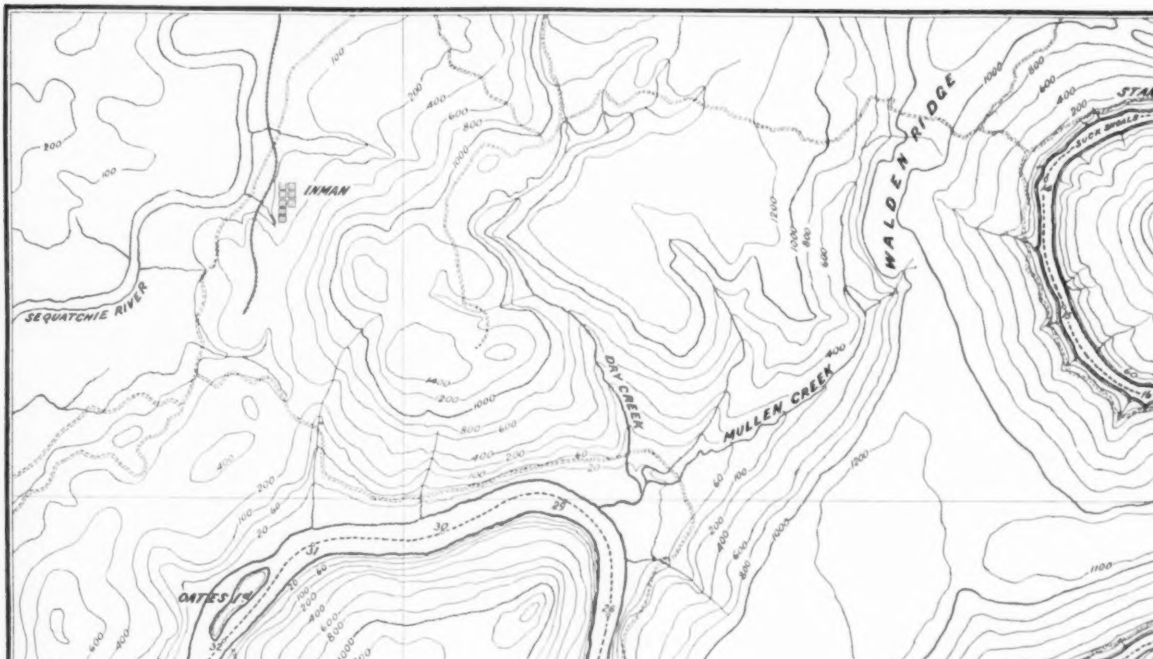
700,000 cubic feet per second and the minimum as 3,700. These results were obtained by extending the discharge curve from intermediate observations to the extreme gauge readings. In 1904 we had the lowest low-water ever known by white men on this river. The lowest preceding one was in 1837, and this was lower than that. An observation taken in that year (1904) gave a discharge of about 4,800 cubic feet per second, taken at the Walnut Street bridge here by the hydrographic observer under the geological survey department. We figure that the ordinary low water will rarely go below 6,000 cubic feet per second. It occurred to me that possibly the members of the Society may be interested to know that Congress, at this session, has passed a law authorizing the development of the water power at Muscle Shoals, where the Government has built a canal to surmount the shoals about fourteen miles long, with a total fall of about 86 feet. I suppose there are various parties who have been figuring upon the development of at least some portion of that fall. The low-water discharge there is greater than it is here; it is probably 9,000 cubic feet there as compared with 6,000 cubic feet here, ordinary low-water discharge.

Mr. Guild.—I might say that there is one feature about this development which may be interesting to you, and that is the great strain upon this dam and the high lift of the gates. I understand that these gates are the highest single-lift lock gates that have ever been proposed in the United States. I am correct about that, I think.

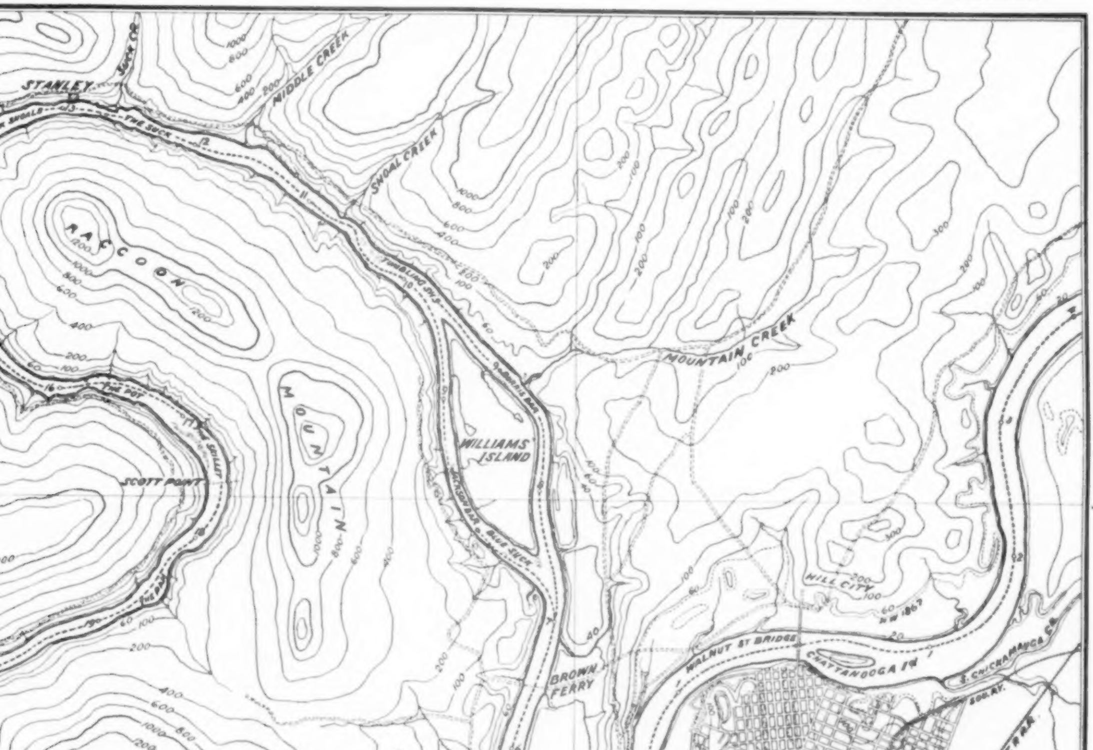
Major Newcomer.—Yes, I think that is so.

Mr. Guild.—And the strain due to the crest of water on the dam is probably greater than on any other. This dam will be 1,200 feet long, with a height of 68 feet, and there will be 22 feet of water flowing over the top which subjects this dam to an enormous strain. I think the figures of the Government Engineering Department show that this dam is subject to a greater strain in this way than any other dam ever constructed in the United States. There have been very much higher dams built, but the water does not flow over them with so great a depth.

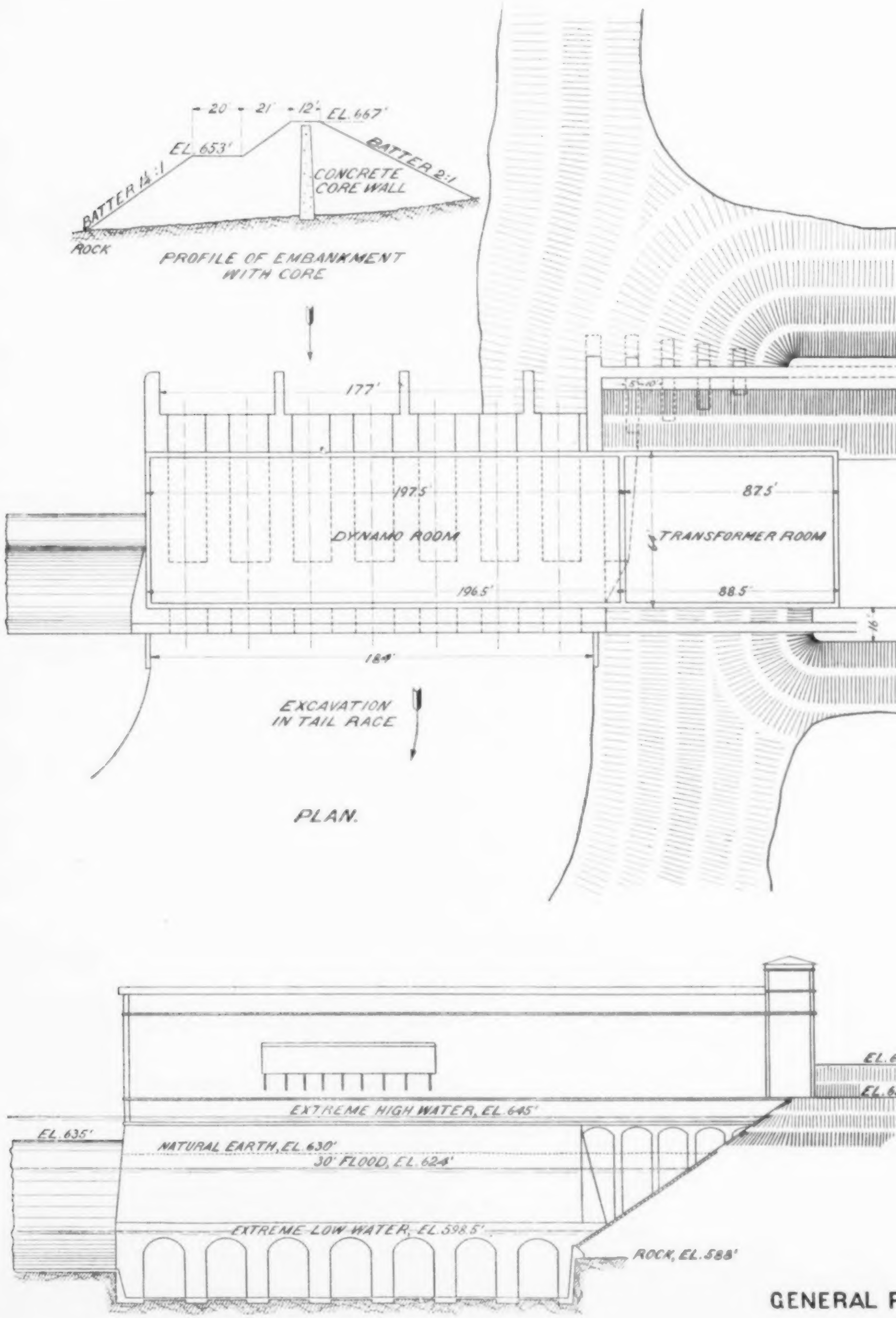
Secretary Hutton.—It is doubtless known to members of the Society that this paper which we have had the honor of listening to has been presented to the Society in two forms. The first or simpler form was the one which was sent out to the members through the Secretary's office. The other form, the more elabo-



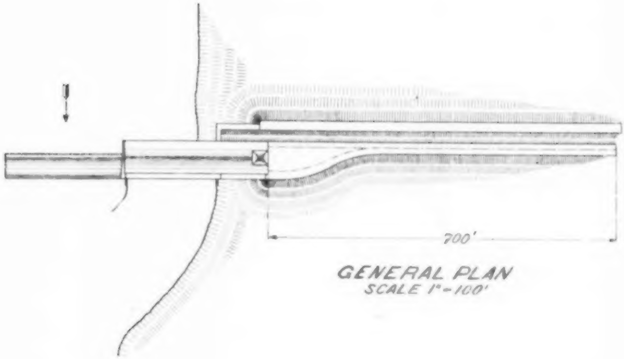
MAP OF "MOUNTAIN SECTION" OF



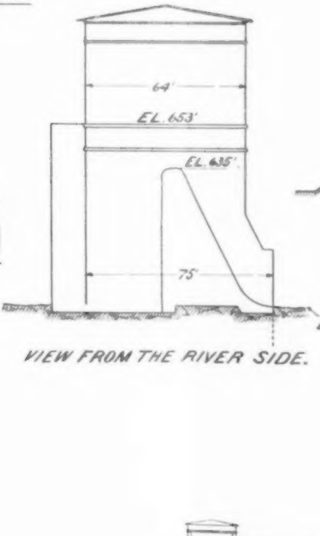
N" of TENNESSEE RIVER.


20'
21'
12'
EL. 667'
EL. 653'
BATTER 1½:1
BATTER 2:1
CONCRETE CORE WALL
ROCK
PROFILE OF EMBANKMENT WITH CORE
177'
197.5'
87.5'
DYNAMO ROOM
TRANSFORMER ROOM
196.5'
88.5'
184'
16'
EXCAVATION IN TAIL RACE
PLAN.
EXTREME HIGH WATER, EL. 645'
EL. 635'
NATURAL EARTH, EL. 630'
30' FLOOD, EL. 624'
EXTREME LOW WATER, EL. 598.5'
ROCK, EL. 588'
VIEW FROM DOWN STREAM SIDE

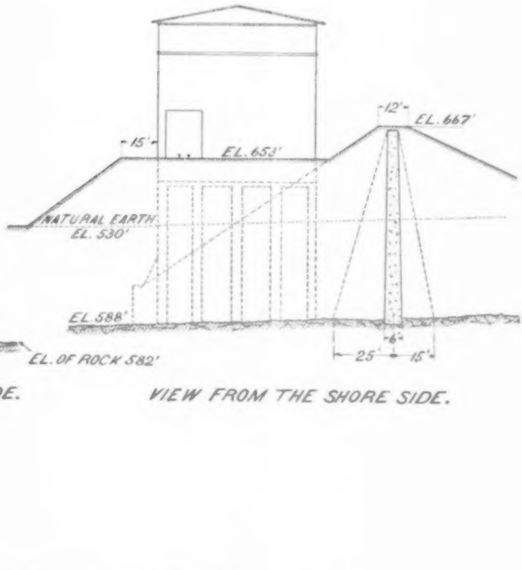
GENERAL P



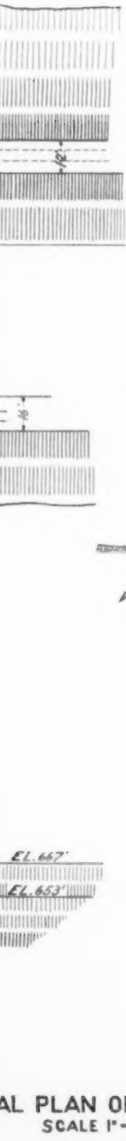
GENERAL PLAN
SCALE 1"-100'

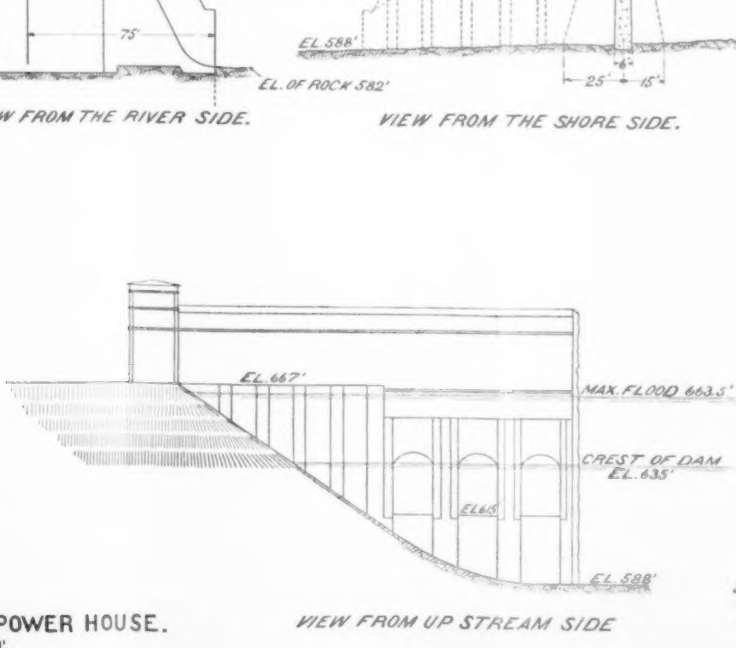


VIEW FROM THE RIVER SIDE.



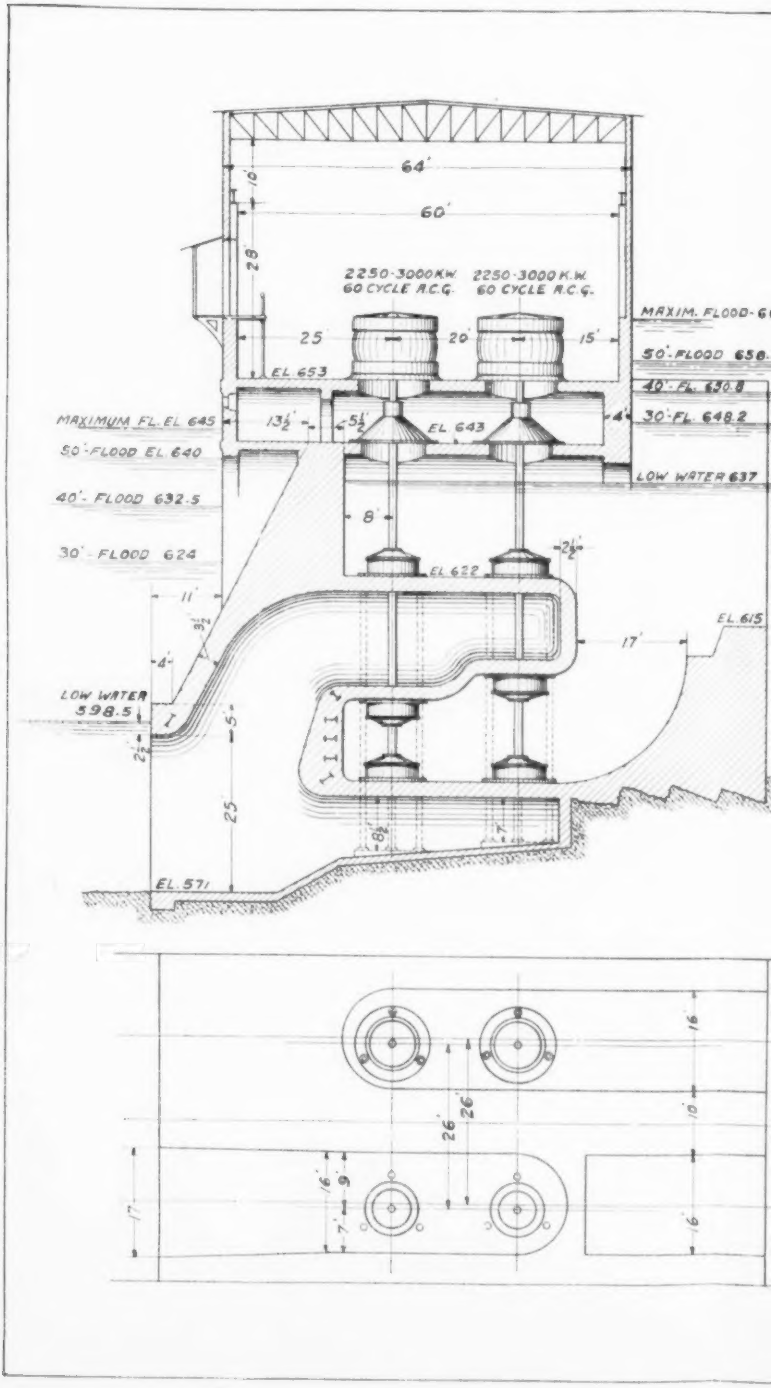
VIEW FROM THE SHORE SIDE.

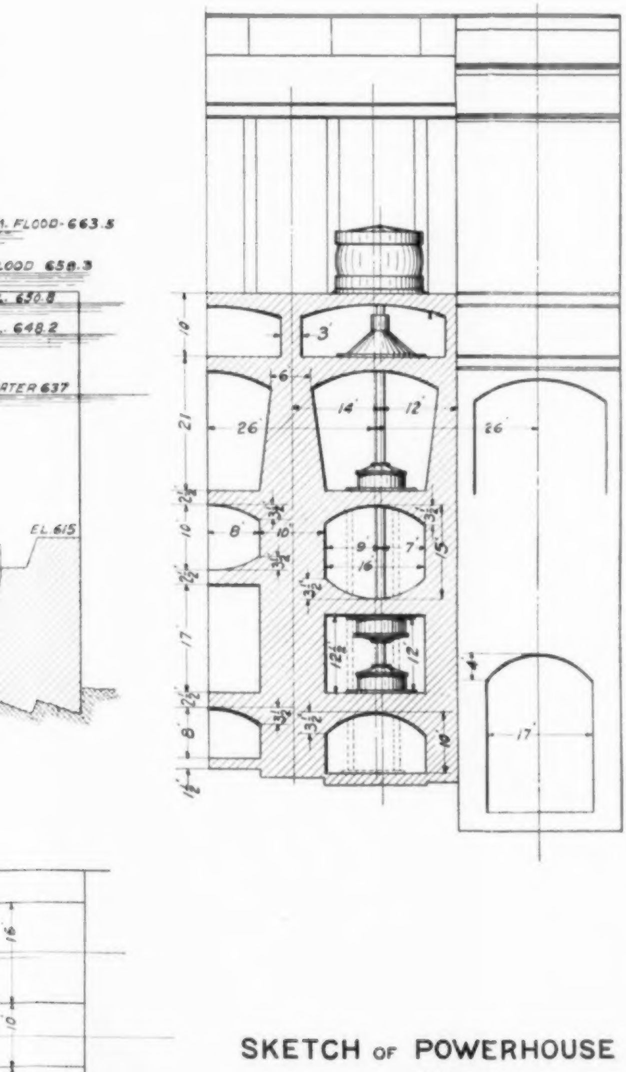




AL PLAN OF POWER HOUSE.
SCALE 1"-20'.

VIEW FROM UP STREAM SIDE


64'
60'
10'
28'
2250-3000 KW 60 CYCLE A.C.G.
2250-3000 K.W. 60 CYCLE A.C.G.
25'
20'
15'
EL. 653
MAXIM. FLOOD-6
50'-FLOOD 658.
40'-FL. 650.8
30'-FL. 648.2
LOW WATER 637
MAXIMUM FL. EL 645
13½'
5½'
EL. 643
4'
50'-FLOOD EL 640
40'-FLOOD 632.5
30'-FLOOD 624
8'
2½'
EL. 622
11'
3½'
4'
17'
EL. 615
LOW WATER 598.5
5'
2½'
25
8½'
7'
EL. 571
16'
10'
26'
26'
16'
9'
7'
17'
16'



SKETCH OF POWERHOUSE AND TURBINE ARRANGEMENT.

rate presentation, was given by the members as they reached Chattanooga. That more complete presentation, with considerable additional information, is a personal gift to the members and guests of the Society by the author of the paper. It seems to me, therefore, very fitting that this souvenir publication containing so much other material that Mr. Murray has gotten together, and which we are pleased to take away with us as a souvenir of the meeting, that inasmuch as we are indebted to him personally for it, and not to anybody else other than he, a vote of thanks would be a very appropriate vote with which to close the discussion on this paper. I might add that I happen to know that this presentation by Mr. Murray represents in value to him $1,200. There are 120 of us here at this meeting. So each one of those books is $10 apiece.

I move you, sir, a vote of thanks to Mr. Murray for his courtesy and the preparation of this souvenir edition. (Applause.)

Mr. Hunt.—I very cheerfully second the motion.

President Taylor.—The motion is carried by acclamation.

No. 1106.*

A LOW RESISTANCE THERMO-ELECTRIC PYROMETER AND COMPENSATOR.†

BY PROFESSOR WM. H. BRISTOL, HOBOKEN, N. J.
(Member of the Society.)

The thermo-electric pyrometer herein described is adapted for commercial and every-day shop use.

It is similar in principle to the Le Chatelier pyrometer, but is of low resistance, and instead of the extremely delicate suspension galvanometer a Weston special dead-beat milli-voltmeter is used, and in place of the costly platinum-rhodium elements, inexpensive alloys are employed for the couples.

The low cost of the couple makes it possible to keep an extra one on hand for use as a standard to quickly and easily check the one that is in regular service.

The temperature at a number of localities may readily be observed on a single instrument, a couple and leads being provided for each locality in connection with a suitable switching device.

The same instrument may also be provided with scales for different total ranges.

For ranges of temperature up to 2,000 degrees Fahr. instead of using porcelain tubes for insulation, each element of the couple is insulated with asbestos and a carborundum paint. Couples so insulated may be applied directly to the fire space where the temperature is to be measured, or where extra protection is desirable the couple may be slipped into a piece of common iron pipe with one

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* Presented at the January, 1906, Reunion in New York City and at the Chattanooga meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

† For further discussion on this topic, consult *Transactions* as follows:

No. 23, vol. 2, p. 42: "Use of the Calorimeter as a Pyrometer for high Temperatures." J. C. Hoadley.

No. 65, vol. 3, p. 187: "Specific Heat of Platinum, and use of the Metal in the Pyrometer." J. C. Hoadley.

No. 874, vol. 22, p. 143: "Recording Air Pyrometer." Wm. H. Bristol.

end closed. Couples so protected are well adapted for use in liquids and molten baths, such as are employed for hardening and tempering of steel.

For instantaneous determination of the temperatures of molten metals as brass, bronze, etc., the ends of the couple are left disconnected and without insulation.

The same form of the couple may be used for quickly measuring the temperature of a metallic object. For this application the tips of the couple are pointed, so that the temperature at the two points of junction may become the same as that of the object immediately after the contact is made. This form of the couple affords a most convenient method for almost simultaneously measuring the temperature at different points of an object.

When desirable the couples are made with a separable junction, which permits the fire-end to be removed and renewed at pleasure.

A compensator is adapted to automatically correct for atmospheric changes of temperature at the cold ends of the couple.

In order to make an equivalent and to reduce the cost of the platinum-rhodium couple for the measurement of temperatures above the fusing point of the low-priced alloys, a compound couple is formed with platinum-rhodium for the part to be exposed to the full temperature to be measured and of a length extending to a point where the temperature will not exceed 1,200 degrees Fahr. The remaining portion of the elements of the couple are composed of inexpensive alloys.

Automatic continuous records of the indications of the pyrometer may readily be made on a chart sheet which is arranged to move at the proper speed back of the end of the indicating arm. This record sheet is unsupported over its active portion, which is periodically vibrated by the clock movement into contact with the end of the indicating arm and produces a record upon the chart sheet.

The record may be made by ink carried by the indicating arm, or the surface of the record sheet may be coated with some easily removable substance.

For automatically recording rapid changes of temperature a current from an induction coil may be passed through the record sheet from the end of the indicating arm at frequent intervals.

1. For a great variety of industrial processes and also in scientific research the ranges of temperature required do not exceed 2,000 degrees Fahr.

2. The low resistance pyrometer herein described has been developed to meet the existing demand for an instrument to fully cover this range of temperature; one which would be accurate, reliable and comparatively inexpensive, taking into consideration both the initial cost and the cost of maintenance; also one that might be readily adapted to varying conditions for industrial operations and be successfully operated by an ordinary workman.

3. As the title implies, this pyrometer depends primarily upon the well-known thermo-electric couple, consisting of two dissimilar metals or alloys joined at one end.

4. When the junction of such a couple is located at a point where the temperature is to be measured an electro-motive force is developed, which is a function of and depends for its value upon the difference of temperature at the junction and that at the opposite or so-called cold ends of the two elements forming the couple.

5. The phenomenon that an electric current is produced when the opposite junctions of two dissimilar metals or alloys are of different temperatures was discovered by Seebeck in 1820.

6. Le Chatelier in his recent book entitled "High Temperature Measurements" states that in the year 1830 Becquerel had the first idea to profit by the Seebeck discovery for temperature measurements.

7. During the seventy-five years since that date many scientists have systematically studied and carried out investigations with thermo-electric couples, using a great variety of metals and alloys, with a view to discover a couple that would resist high temperatures and could be depended upon for constancy when used for their measuremnt. Le Chatelier, who has made extensive researches to determine the most desirable metals for this purpose, finally adopted a couple of which one element consisted of pure platinum and the other of an alloy of platinum and 10 per cent. rhodium, from which couple an almost uniformly increasing electro-motive force is developed, correspondingly with increasing differences of temperature between its opposite ends.

8. At the present time many of these couples are successfully employed for the determination of high temperatures. They are almost invariably used in conjunction with an extremely delicate high resistance galvanometer, which, according to Le Chatelier, is indispensable, 200 ohms being mentioned as a minimum resistance allowable in the indicating instrument. This amount of resistance

is necessary to practically eliminate the atmospheric temperature influence upon the resistance of the elements forming the couple, the leads and the coils of the galvanometer itself.

9. If the galvanometer be of a high resistance type it is evident, that if sensitive to the minute current of electricity corresponding to the very low electro-motive force produced by the couple, it must be delicately constructed, and consequently requires great skill and care in its handling and operation.

10. The sensitive coil of the instrument is usually suspended by an extremely fine wire and the instrument must always be leveled and located upon a solid foundation before observations can be made.

11. The platinum-rhodium couple instrument above described may be classed as a high resistance pyrometer when compared with the *low-resistance pyrometer,* of which the following is a description:

12. It consists of three parts: couple, indicator and leads to connect couple and indicator. The leads between the couple and the indicator may be of almost any desired length to meet the special requirements; the combined resistance of the leads, couples and indicator are fixed to suit the total range of the instrument and varies from 3 ohms as a minimum to 10 as a maximum. The indicator is a low-resistance instrument of special design, and is made especially for the writer by the Weston Electrical Instrument Company. The accuracy, permanency, and portability of these instruments is well known.

13. Fig. 1 shows a wall or switch-board form of the indicator. Fig. 2 illustrates a portable form. These indicators are made with pivots in jeweled bearings in place of the delicate suspension by fine wires which are generally considered necessary in the high-resistance type of indicator or galvanometer for this work.

14. The elements used in the low-resistance system give a much greater electro-motive force than the platinum rhodium couple, which is a great advantage in gain of motive power for the operation of the indicating instrument.

15. The particular metals or alloys applicable for a pyrometer of this type should have a fusing point higher than the maximum temperature to be measured, and when formed into a couple should produce a high electro-motive force with practically uniform increase of same proportional to the increase of temperature.

16. As the result of many experiments with different metals and

alloys to determine suitable materials to meet these requirements, couples have been finally adopted which consist of alloys of tungsten, steel, nickel, iron and copper; different alloys being employed to suit the total ranges of temperature that it is desired to have the scale cover.

17. Since no rare metals are used for the couples, this part of the pyrometer is inexpensive, and it is possible to employ elements of large cross-section, which will not be affected in their resistance

FIG. 1.

any appreciable amount by the variation of temperature along the lengths of the elements forming the couple.

18. The leads to the indicator are made of flexible insulated copper duplex cable, and of ample cross-section to practically eliminate the influence of variations of atmospheric temperature.

19. The cross-section of the elements of the couple is reduced at their junction, thus rendering it sensitive to sudden changes of the temperature to be measured.

20. A novel feature of the couple is that it is made separable at the point where it passess through the wall of the space within which the temperature is to be measured.

21. The object of the joint is two-fold: first, to make it possible to renew the "Fire-end" whenever it may be necessary; and second, to permit carrying the cold ends of the elements to a point toward the floor where the atmospheric temperature will be constant and not influenced by the temperature that is being measured.

22. Fig. 3 shows a complete element with separable joint, coiled leads and lamp plug on the end of the cable forming the leads for convenient connection to the indicating instrument shown in Fig. 1.

23. A special feature of the joint is that it is provided with

FIG. 2.

large bearing surfaces to prevent possibilities of variations of resistance at the connection and is constructed to allow for easily breaking and making the connection. The details of the joint are shown clearly in Fig. 4, and from this it will be seen that it is impossible to make the connection incorrectly, as is usually done when there is nothing to guard against it.

24. The low cost of the couples makes it practicable for the user of the instrument to keep an extra fire end in reserve, which may, at any time, be quickly substituted for the one that has been in continual service, thus affording an economical and positive check upon the accuracy of the instrument.

25. The elements of the couple are independently insulated in a novel and effective manner by winding each with asbestos cord and then coating the surface with carborundum paint, a solution of silicate of soda being used as a binder. This makes a clean, compact and smooth insulation.

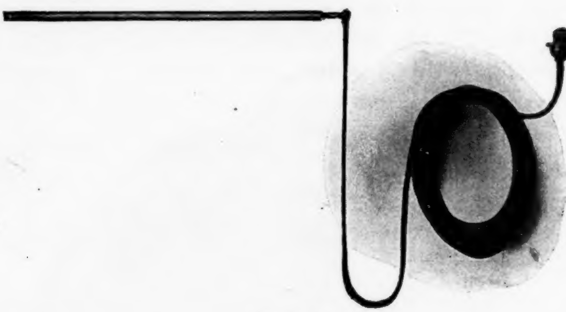


FIG. 3.

26. Couples thus insulated are flexible, and can either be applied to the heated space directly or they may be inserted into a piece of ordinary pipe as a protection. This protection has proven itself to be effective and economical.

27. For continuous applications of the couples to temperatures

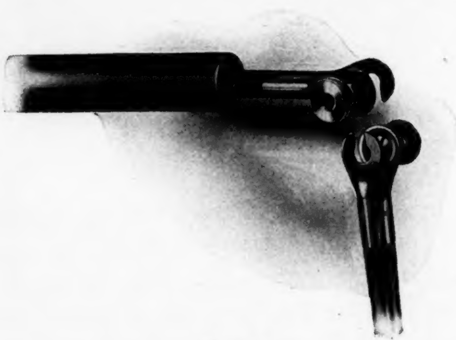


FIG. 4.

in the neighborhood of 2,000 degrees Fahr. or over, special protecting tubes of nickel, plumbago or porcelain are employed.

28. These pyrometers are furnished with scales for total ranges of 600, 1,200, 2,000 and 2,600 degrees Fahr. Reproductions of these scales are shown in Figs. 5, 6, 7 and 8.

29. The graduations of these scales are determined by the fusing temperatures of lead, zinc, aluminum and copper, which give sufficient points on the curves for use in making a complete graduation of the scales.

The divisions are also further checked by the use of a standardized Le Chatelier Platinum-Rhodium Couple, and a Sieman-Halske Suspension Galvanometer.

For very open scales over shorter ranges several couples may be placed in series, thus making it possible to read to small fractions of a degree.

30. A novel application of the thermo-electric couple is that of determining the temperature of molten metals, as cast iron, copper, brass and bronze. It consists in leaving the ends of the elements disconnected and without insulation. When these ends are slightly immersed in the molten metal it makes a junction between the elements and the reading will be same as if the elements of the couple had been originally joined. The advantage of this plan is that the reduced cross-section at the ends of the couple allows it to almost instantaneously attain the temperature of the molten metal, consequently there is no lag error —a most important advantage. As the couple is used over and over again, the ends become worn away, but the couple is, nevertheless, always ready for use by immersion of a fresh portion, which will not be changed in any way by continued use and will give the same reading for a given temperature as if the couple had not been worn away. A joint is provided near the end that is immersed so that a fresh tip can be applied to the couple before enough of the end has worn away to appreciably affect the resistance of the complete system.

Compensators.

31. As already mentioned, it is well known that the electromotive force generated by a thermo-electric couple is a function of the temperatures at the hot and cold ends.

32. For refined measurements it is therefore necessary to make allowance for changes of temperature at the cold ends of the couple when readings are taken, unless some means is provided to maintain them at a constant temperature. This is sometimes done by immersing the cold ends in ice water or by having a water jacket around the ends through which there is a flow of water at some known temperature.

33. In the Low Resistance Thermo-Electric Pyrometer System, comparatively small changes in the actual resistance of the circuit including couple, leads and instrument will produce sufficient effect to correct for atmospheric changes at their cold ends.

34. A compensating device to automatically correct for changes of atmospheric temperatures at their cold ends has been devised which makes it possible to dispense with the cumbersome means for maintaining the cold ends at constant temperature or the necessity of taking readings of temperatures at the cold ends.

35. This compensating device is shown in Fig. 5. It consists of a glass bulb with a short stem, similar to an ordinary mercurial thermometer. Two platinum terminal wires are fused into the stem near its top. These are connected within the bore of the stem by a loop of fine platinum wire, thus completing the circuit as indicated in the diagram. The size of the bulb, the cross-section

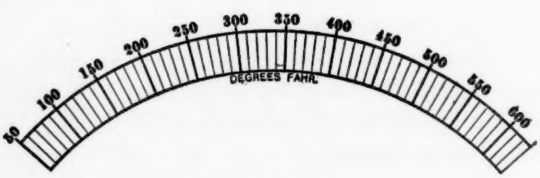


FIG. 5.

of the bore of the stem, and the cross-section of the platinum wire loop are proportioned to suit the case in hand.

36. The compensator will perfectly compensate for any particular point on the scale; as, for instance, the working point, where it may be desired that the reading shall be absolutely independent of changes of temperature at the cold ends. It will readily be seen that if the temperature rises at the cold ends the mercury rising in the stem will short-circuit a certain portion of the platinum loop, thus reducing the resistance of the entire circuit by exactly the necessary amount so that the diminished electro-motive force of the couple due to the rise of temperature of the cold end will send the same amount of current through the circuit and instrument, and consequently give the same reading as if there had been no change of temperature at the cold ends.

37. The compensator acts on precisely the same principle, but in a reverse manner, when the temperature falls at the cold ends, the resistance of the circuit being increased as the column of mercury lowers in the stem. The increase produced in the resistance

of the circuit prevents the increased electro-motive force of the couple due to the fall of the temperature at the cold ends from sending an increased current through the instrument, therefore the reading remains unchanged. The compensator may also be employed within the indicating instrument to correct for atmospheric changes of temperature upon the instrument itself where extremely accurate results are required.

38. It will be seen that a compensator of the form described cannot as conveniently and practically be applied to a high resistance indicating instrument.

General Advantages of Thermo-Electric Pyrometers.

39. As compared with other forms of apparatus for measurement of high temperatures, the thermo-electric pyrometer has many advantages, of which the following are the most important:

They may be employed where the space is extremely small and inaccessible. They are practically independent of temperature variations intermediate of their hot and cold ends. They are independent of pressure and rough usage at the point where the temperature is desired to be measured.

40. The indicating instrument can be located at the most convenient point, practically at almost any distance from the couple. They are extremely sensitive to changes of temperature and respond instantaneously—that is, there is no lag error. They are constant in their indications when the couples are properly protected. They permit the determination of the temperature at many different points by means of several couples and leads connected to one instrument, provided with suitable switching device.

Special Advantages of Low Resistance System.

41. The important advantages of the Low Resistance of Thermo-Electric Pyrometer System may be summarized as follows:

First. A commercial switchboard or portable dead-beat indicating instrument may be employed instead of the extremely delicate suspension galvanometer required for use with a single platinum-rhodium couple. This advantage is gained by the fact already stated, that the thermo-electric couples employed give several times as much electro-motive force as the platinum-rhodium couples, which is ample to successfully operate a pivot instrument if of sufficiently low resistance.

Second. It affords a practical method for automatically compensating for the changes of temperature at the cold ends of the couple, as already described.

Third. It makes it practicable to use the same indicating instrument and the same couple for different total ranges of temperature by using different binding posts and having several scales drawn, the proper resistances being inserted for each individual total scale.

Fourth. The application of low-priced metals and alloys as a substitute for platinum and rhodium makes it possible to install a number of couples, and by means of proper switching devices use an instrument for quickly determining the temperatures at the locations of the different couples. In many instances the first cost of the expensive platinum elements prohibit their use in this way.

Applications.

42. Many applications of this instrument will suggest themselves. A few of the important ones are mentioned, as, for exexample, in a boiler test when nineteen couples were simultane-

FIG. 6.

ously applied at different points between the furnace and the flue. From the data obtained a curve was drawn showing the temperatures at all points along the path of the products of combustion from the furnace to the flue, the abscissas corresponding to the square feet of heating surface, and the ordinates to degrees Fahrenheit. The value of such data for investigating and studying the economical working of steam-power plants will be appreciated. The couple can readily be applied to the steam space of a boiler and used to show the degree of super-heating.

43. These instruments have also been adapted to and are especially valuable in maintaining the desired temperatures for annealing, hardening, tempering and blueing of steel.

44. When many small parts are handled, as in the manufacture of watches, a practical method of using the pyrometer is to

adapt the pot containing the articles to the end of the couple and use it as a handle for inserting the pot into the furnace or into an ordinary forge fire. By revolving the pot it is heated perfectly uniformly, and as soon as the proper temperature is reached it is known on the indicator, and all guesswork is eliminated.

45. They have been most successfully employed in lead-hardening baths. For this purpose the couples are protected by wrought-iron pipes, and will last for months without renewal. After constant daily use for many weeks the couples give the same readings as when first installed.

46. In addition to making it possible to obtain absolutely uniform results with a given lot of steel it has been found that the life of the pots have been increased, as they are not overheated. From this fact it naturally follows there must be an economy of fuel as a result of using the pyrometer.

47. Application has also been made by galvanizing baths, afford-

FIG. 7.

ing means for keeping the molten metal at the proper temperature for the work, preventing overheating and wasting zinc by vaporization.

48. They have been used to keep molten lead at correct temperature in the manufacture of shot.

49. These instruments have also been tried and are now being tested for indicating the temperature in the carbureter and superheater of a Lowe Water Gas Plant.

50. Where the process of gas making depends upon proper temperatures and the pyrometer shows when the steam and oil should be turned off and the blast turned on, and also when these operations should be reversed in order to obtain the most efficient results.

51. By using two or three couples in the carbureter it is possible to adjust the spray of oil so that every part from center to shell will be working to the best advantage.

52. The instruments have also been successfully employed in chemical works.

53. The field of usefulness seems to be very broad, as the instrument can be adapted to meet almost any individual requirement.

Discussion.

Mr. Gus C. Henning.—The apparatus described and shown by Professor Bristol is certainly very ingenious, and the most practicable thus far developed.

The promptness with which changes of temperature are indicated is marvelous, and not obtained by any other apparatus used for the same purpose.

The comparative cheapness is also another good point, as it can thus be used very generally.

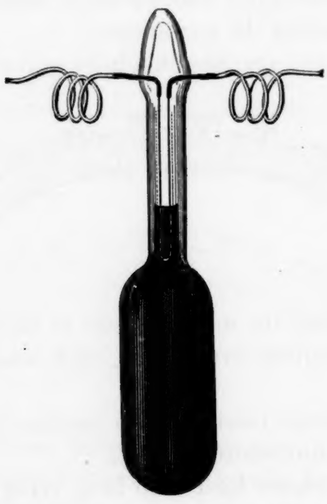


Fig. 8.

The simplicity and cheapness of the couple is also a very advantageous matter, and its interchangeability is unique.

The fact that temperatures can be correctly and continuously recorded is a great advantage.

With this apparatus the expert heater is no longer the most important personage in a works, and cannot play the tyrant as is so commonly the case. Once having determined proper temperatures in furnaces for different classes of work, these can always be reproduced and maintained, and it is no longer a matter of guesswork.

Moreover, one such apparatus with a switchboard makes it applicable to a great many furnaces at the same time.

Prof. Ira H. Woolson.—How nearly would the couple check on the melting point of copper or gold?

Prof. Wm. H. Bristol.—Exactly. Since I use the fusing point of copper to determine that point on the scale.

Mr. Thompson.—In the couples in which you used an alloy of iron, when you pass the transformation point of the iron alloy, does the rate of increase per degree C. of the electro-motive force change?

Professor Bristol.—I have not tried that experiment.

Mr. Henry Souther.—After some twelve years' experiment with pyrometers, and after numerous attempts to introduce them practically into commercial surroundings, I appreciate Professor Bristol's remarks very much. It seems to me his pyrometer smooths out a great many of the rough places, and that it is of a more practical nature than the Le Chatelier type, with its necessarily delicate galvanometer. Also the terminals of the platinum and platinum-iridium or rhodium thermo-couple of the Le Chatelier type become very friable with use. Professor Bristol's couple seems to overcome this objection. The difficulty has always been in introducing the Le Chatelier thermo-couple that every once in a while it failed, and the men using it became disgusted and sidetracked the instrument.

The difficulties in the way of the use of the pyrometer have been practical, and not inherent in the pyrometer as an instrument. The necessary galvanometer is essentially a laboratory instrument, the thermo-couple is delicate and requires nursing. These objections are very serious about a shop.

The use of a pyrometer in a furnace where there are hot and cold draughts is always unsatisfactory, and I believe always will be. If the pyrometer is to be used in a furnace it should actually be buried among the parts or in the material being heated, and not subjected to the draughts of a furnace.

The ideal use for a pyrometer is in connection with a molten bath of lead, cyanide, salt or whatever may be convenient. The temperature of such a bath can be perfectly measured, and the material immersed in it acquires the same temperature if left there long enough. The personal equation of the operator disappears almost entirely, which is not the case nor can it be in connection with a furnace where the couple is suspended over or near the work, and not actually in contact with it.

Professor Bristol's thermo-couple will stand a lot of abuse, and, better yet, will stand immersion in molten baths for a considerable length of time. His electrical instrument is not of the delicate type peculiar to the high resistance platinum, platinum-rhodium couple.

It is for these practical reasons that I believe the Bristol instrument is more acceptable than any other type presented at this time.

Mr. R. L. Penney.—In manufacturing firearms it becomes necessary in many cases to be able to determine quickly the temperatures of baths, furnaces, etc.

The first kind of pyrometer used was one made by one of the well-known companies, but we soon found this kind to be as vari-

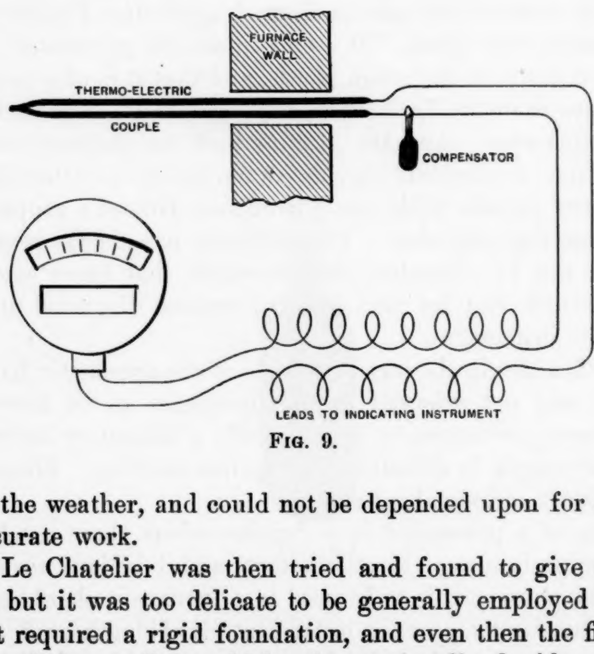


FIG. 9.

able as the weather, and could not be depended upon for anything like accurate work.

The Le Chatelier was then tried and found to give excellent results, but it was too delicate to be generally employed for shop use. It required a rigid foundation, and even then the fiber used to suspend the needle would break occasionally, besides trying to keep the temperature of the cold junction constant or at zero degree C. was very difficult.

Beginning with last June, we have introduced the William H. Bristol Thermo-electric Pyrometer, and at present we are using it in four (4) different branches of our work, hardening, tempering, blueing and annealing, also using it in place of the Le Chatelier for experimental work. The results obtained so far leave

very little to be desired. The instruments are arranged so that the workman knows just what he is doing, and the element of doubt and uncertainty is eliminated.

The hardening furnaces each have a couple, a compensator, and a set of leads, while one instrument does for all. The instrument is placed on the wall in a place convenient for all the workmen; near it is a small switchboard containing a switch for each furnace, the switch having a common connection to the instrument. The leads are brought from the several furnaces to the individual switches. The furnaces and switches are numbered to correspond. By this means a workman who, we will say, is working on No. 4, is required to keep his furnace at a certain temperature. He goes to the switchboard and throws in No. 4 and reads the temperature directly, without making any correction, without waiting for an oscillating needle to come to rest. The couples are introduced into these furnaces from the back, the fire-ends being enclosed in a porcelain tube. These couples have required no attention since they were set up.

The same scheme is applied to our tempering and blueing baths, using a lower range instrument. In these last two applications accuracy and permanency are very essential points.

With these pyrometers as now in use we are enabled to obtain results that are unchanging from day to day.

In annealing these instruments and couples are of great benefit. On account of the inexpensiveness of these couples we use two for each furnace, one to measure the temperature of the furnace and another to measure the temperature at the center of the box containing the work to be annealed. The leads from each couple are carried to a double-throw switch. The workman in charge throws the switch one way to find the temperature of the furnace, and the other way to find the temperature of the work.

The two couples are introduced together through the side of the furnace, one reaching into the furnace a short distance, and the other extending through the fire space into the center of the box through a hole in its side. This arrangement has been found to be satisfactory and enables the workman in charge to handle the furnaces as if they were machines, slacking and starting them up as necessary.

Having compared the Bristol pyrometer with the Le Chatelier and found it accurate, we have used it for experimental work in many cases, of which the following is an example: Wishing to de-

termine the rate of the rise of heat in a case of cartridges and the temperature at which they would explode when heated gradually from the outside, two couples were introduced into the case, one a short distance and the other into the center. The instrument was about 90 feet distant, and heat was applied to the outside of the case which was enclosed in a sheet-iron oven. The heating continued until all the cartridges had exploded. The cartridges exploded one by one, there being no general explosion. On examining the couples after the explosion they were found to be bent and twisted out of shape, but no indication of this was shown on the instrument during the experiment. One looking at the instrument would not guess that the couples were being used so roughly. The temperature rose steadily to a maximum and then dropped. If it had been necessary to use two platinum rhodium couples this would have been an expensive and difficult matter.

We have found the Bristol pyrometer to be one that can be applied to every-day shop problems successfully.

Prof. Wm. Kent.—The accurate measurement of high temperature measurements is a problem of ever-increasing importance to the industrial arts. It is greatly to be desired that those who are trying to improve the commercial methods of measuring high temperatures will be able to produce an instrument by which any temperature up to 3,500 degrees Fahr. may be read by a workman as easily as he can read the pressure of steam in a steam gauge, and that these instruments shall also be reasonably durable under sudden alternations of temperature, which the pyrometers using porcelain tubes are not.

It is well known that in steam boiler practice the average economy of any boiler ordinarily is at least ten per cent. lower than the same boiler is capable of when all the conditions of running are exactly what they should be. If the boiler is not overdriven, the variable condition which influences economy the most is the amount of air supply per pound of fuel. If this can be kept down to from 18 to 22 pounds of air per pound of carbon, the economy will be a maximum, but if it is increased to 30 or 40 pounds, as very often happens in ordinary boiler practice, the economy will fall off rapidly, especially at high rates of driving. The trouble is that the ordinary fireman has no means of knowing when his air supply is just right. He has two means for controlling it, (1) regulating the damper, and (2) varying the thickness of the bed of fuel, but he cannot vary it intelligently, for he has no

means of measuring or indicating the air supply. It can be indicated by a pyrometer; for the less the air supply, provided it is not so small as to make imperfect combustion (which is not likely to take place below 18 pounds), the higher is the temperature. It is possible with almost any kind of coal containing not over two per cent. of moisture to obtain a furnace temperature, in a furnace roofed over with a fire-brick arch, of 3,000 degrees Fahr., and this temperature can be maintained if the firing is done in small quantities at frequent intervals. If the fireman had a pyrometer which would indicate just what his furnace temperature was, he could so adjust the thickness of the fire and the force of the draft to maintain that temperature, and thus obtain maximum economy. If such a pyrometer could be made with a registering attachment, so that the superintendent of a plant might at times know what temperatures have been maintained in a furnace, it would lead to a great increase in economy of fuel in steam boiler plants.

Several years ago I had an opportunity to make a long series of boiler tests with different fuels, and in these tests I used a Uehling and Steinbart pyrometer to indicate and to record continuously the furnace temperatures. With Pittsburg coal I was able to hold the temperature always between 3,000 and 3,300 degrees Fahr. With coals higher in moisture it was impossible to reach 3,000 degrees, but such temperatures as could be reached were maintained within a range of 200 and 300 degrees by means of firing every five minutes. With longer intervals of firing the temperatures varied more. It is not to be expected that this pyrometer can be regularly used in steam boiler practice, as it is rather expensive, and requires some skill in handling it and keeping it in order, but some pyrometer that will do the work which it did on that occasion is greatly to be desired.

Another branch of industry in which measurement of high temperatures may prove to be of considerable importance is the welding of steel pipe. A series of tests made by Mr. T. N. Thomson, of the International Correspondence Schools, and reported recently to the American Society of Heating and Ventilating Engineers, shows that the strength of the weld in an apparently good steel pipe may vary anywhere from 50 per cent. to 90 per cent. of the strength of the steel itself. It is highly probable that this difference in the strength of welds is due to a difference in the temperature of the furnace in which the steel is heated, and if so, the use of a pyrometer would help in the discovery of the proper tem-

perature, and also be of assistance to the men in charge of the furnace to enable them to know just when the temperature was right.

In the heating, tempering and annealing of steel the question of temperature is all important, and much more uniform results could be obtained if pyrometers were used by the workmen in charge of these several heating operations.

Mr. Albert A. Cary.—There is probably no field of scientific research opening the way to more useful information in the manufacturing and chemical arts than careful and accurate observations of the effect of temperatures upon various substances, and also the thermal study of furnaces required to effect these heat actions.

In days gone by, and not at very remote dates either, we have heard of *wonderful* results obtained by experimenters treating certain substances, or combinations of materials, by heat; such remarkable results being obtained accidentally. Weeks and months of careful effort have often followed such accidental discoveries to reproduce them, but without avail.

It is safe to assert that in most of these cases an accurate knowledge of the action of heat upon these substances would have helped the experimenter to reproduce these results in a very short time. To illustrate roughly the importance of this assertion, I will refer to certain thermal information necessary to alloy work.

Gold and copper have been frequently used together to form useful metal mixtures, and for a long time, previously to a comparatively recent date, it was not known whether the fusion or freezing point of copper was higher or lower than gold. The fusion and freezing point of gold was determined quite accurately at 1,949 degrees Fahr., but it was not until after carefully constructed high temperature measuring and recording apparatus was used that the uncertainty concerning the freezing point of copper was settled, when it was found to have two freezing points, viz., 1,949 degrees Fahr., and 1,983 degrees Fahr.

The higher temperature was obtained by use of pure metal, melted in a graphite crucible, protected from the air on its upper surface by a layer of powdered graphite.

The lower temperature was obtained by melting the copper exposed to the oxidizing effect of the air, and in the course of ordinary practice all sorts of freezing temperatures are obtained between these two limiting degrees.

The cooling curve of copper is shown in the cross-section plot marked A, and was obtained by immersing the end of a thermo-

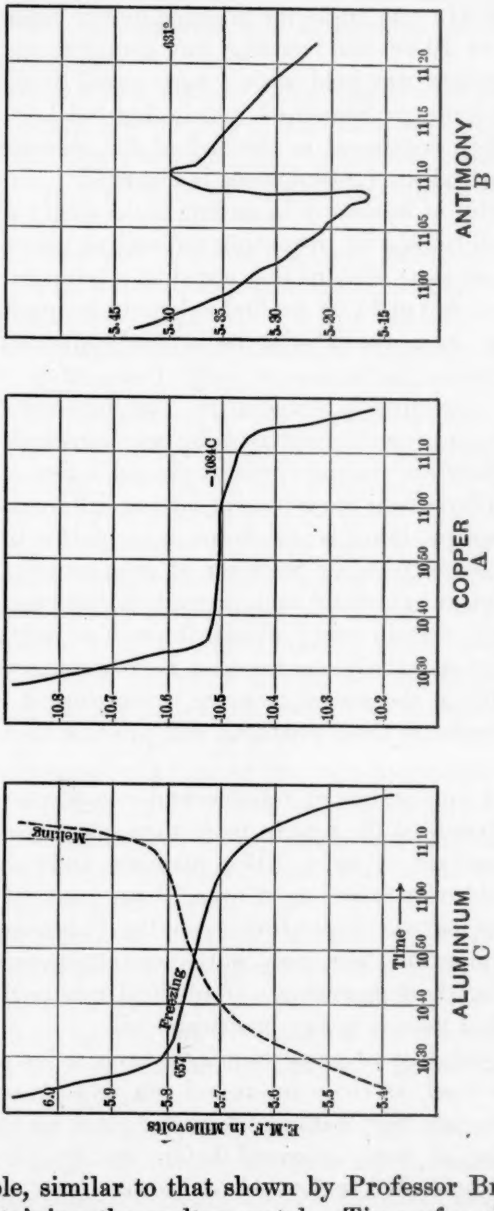


FIG. 10.—COOLING CURVES.

electric couple, similar to that shown by Professor Bristol, in the crucible containing the molten metal. Time of cooling is here

plotted as abscissa, and the E. M. F. of a platinum and platinum plus 10 per cent. rhodium thermo-couple, is plotted as ordinate.

This shows the molten metal heated to a temperature of 2,048 degrees at 10-27. At 10-33 the freezing action began to affect the metal, and at 10-36 the freezing was complete at 1,983 Fahr. This temperature was held with a very small drop, until 10-58, when a slow cooling commenced, continuing until 11-14, and then a rapid cooling continued to the end of the observation, when a temperature of about 1,940 degrees is observed.

The behavior of antimony in cooling is decidedly different from copper, as can be seen by inspecting the cooling curve in plot B.

At 10-51 we start with molten metal at a temperature of 1,179 degrees Fahr., and at 11-08 we find a drop in temperature to 1,112 degrees Fahr., when the freezing begins and continues until 11-9½, when the freezing is complete at 1,168 Fahr., after which follows a somewhat rapid drop in temperature, and then a slower fall.

As aluminum presents entirely different characteristics in freezing, I will show the cooling curve in the solid line of plot C, and this metal differs from the others in having different freezing and melting points, the latter being shown in the dotted line.

Aside from the peculiar behavior of such metals, subjected to heat treatment, it has been found, through delicate and careful heat measurements, that in many chemical products certain reactions and combinations take place between narrow ranges of temperature, and without the use of accurate pyrometers it would be impossible to duplicate these products, and produce them uniform in quality.

The recent appearance of reliable high temperature measuring devices has benefited the arts in more than one direction.

A manufacturer of twist drills, reamers, and other tempered steel tools told of a rejection of over 10 per cent. of his finished product when he was depending upon the judgment of his employees in tempering these tools, but since introducing pyrometers for controlling the temperatures of his heat treatments, his rejection of finished tools is to-day practically nil.

There is probably no metal requiring such refinements in heat treatment as steel, as those interested can find by referring to a lecture by the late Sir William Roberts-Austin on the hardening and tempering of steel, delivered before the British Association in 1889, and published in Vol. 41 of Nature.

Generally speaking, the state of pyrometry in this country, as

applied to manufacturing and chemical industries, is, even at the present time, in a deplorable condition, and many of the instruments which we find in use are obsolete and unreliable, and incapable of leading to improvements in methods of manipulation or in refining products.

The most accurate temperature measuring apparatus in common use is the mercurial thermometer. The first limit found to measurements of high temperatures with thermometers was due to the comparatively low boiling point of mercury, which, under atmospheric pressure, is 676 degrees Fahr.

As thermometers have been constructed, until recently, with a vacuum over the mercury in the stem, the boiling point is reduced to a lower temperature than 676 degrees, and 550 degrees has been their limit of indication. Generally speaking, all references to instruments which measure temperatures above this degree have been termed high temperature measuring apparatus.

In order to raise the boiling point of mercury, thermometer manufacturers began by introducing air under more or less pressure above the mercury, and then sealing the tube. Two troubles now appeared—first, the mercury began to oxidize at higher temperatures, and then the glass began to soften and alter in its dimensions.

The introduction above the mercury of nitrogen or carbonic acid under pressure did away with the oxidizing effect, and better glass was produced, which decreases greatly the expansion with permanent set, which led to what is known as a depression of the zero.

A glass now used by a few of the best thermometer manufacturers, known as Jena 59 III., is a borosilicate glass which, when used for thermometers, has a very small zero depression, and if properly treated the thermometer made from it can be used safely at a temperature of 1,000 degrees Fahr.

This thermometer, when treated, is annealed by being placed in a furnace maintained at a temperature of at least 1,200 degrees Fahr. for not less than 75 hours, and then allowed to cool very slowly. Such an instrument will indicate a temperature of 1,000 degrees Fahr. for long intervals of time, with no material variation or depreciation of its zero.

This mercurial pyrometer is preferably filled above the mercury with nitrogen under 60 pounds pressure per square inch, and on account of the nitrogen pressure I have found that in general work, where required, it can be used upside down with no appreciable error.

The limit in temperature measurement in the thermometer thus constructed is due to the softening of the glass, and this led to the investigation of quartz for an encasing material, quartz having a fusing temperature of over 4,000 degrees Fahr.

Owing to the excellent work done by Heraeus, of Hanau, and Siebert & Kuhn, of Cassel, in the manipulation of quartz, mercurial quartz thermometers, nitrogen filled (under very high pressure) can be obtained which will safely indicate temperatures of 1,300 degrees Fahr. These are necessarily very expensive, but owing to great toughness of the quartz they are not easily broken and are very durable.

Pyrometers of this construction reading to much higher temperatures have been attempted by Dufour in France, using tin in place of mercury for the contained liquid, which liquid now becomes the heat-limiting material, but I understand that only a fair degree of success has attended his efforts.

There is sometimes a material error attached to the use of mercurial thermometers in measuring high temperatures, due to there being only a fractional part of the stem immersed in the hot bath, the balance projecting into the cooler air. Under such conditions, with several hundred degrees temperature being measured, the cold upper part of the stem will cause too low a reading.

C. W. Waidner, of the United States Bureau of Standards, has offered a formula for correcting this error, which I have found from tests with my calibrating apparatus to be quite satisfactory.

This formula is as follows:

$$\text{stem correction} = 0.00016\, n\, (T—t) \text{ degrees Cent.}$$

and

$$\text{stem correction} = 0.000088\, n\, (T—t) \text{ degrees Fahr.}$$

in which

n = Number of degrees emergent from the bath.

T = The temperature of the bath.

t = Mean temperature of the emergent column.

These corrections may amount to 45 degrees Fahr. at very high temperatures.

Above 1,000 degrees Fahr., we are now driven to the use of some other form of temperature apparatus, and I have found nothing more satisfactory and reliable than the thermo-electric pyrometer in measuring temperatures up to 2,900 degrees.

The instrument of this type such as I have been using is known

as the Modified Le Chatelier Thermo-electric Pyrometer, and differs from that shown by Professor Bristol in being high in cost, requiring some degree of delicacy in manipulation as well as experience in application, but it is certainly a very sensitive measuring instrument and accurate beyond question, as I have proved again and again by calibration.

The thermo-electric pyrometer is by no means a new apparatus. It was first proposed by Becquerel in 1826, but not used scientifically until ten years later by Pouillet, who had but fair success in its application.

C. Barus did much valuable work in the development of this form of pyrometer, in connection with his work when attached to the United States Geological Survey, when he investigated the temperature at which the various igneous rocks were formed; while Le Chatelier did an immense amount of work in connection with the thermo-electric as well as the optical pyrometer, and he introduced the present most widely used couple, composed on one side of pure platinum, and on the other side of platinum alloyed with 10 per cent. of rhodium, and that was somewhat over twenty years ago.

The principle of the thermo-electrical couple is simple. It consists of two metals joined together at one end, and when heated at this juncture a weak electric current is established which flows through the wires to a delicate galvanometer, or preferably a volt meter, which will read to 1-1000 of a volt. The electro-motive force of this current is a function of the temperature which generates the current, so by noting the voltage and by comparing this reading with a curve formed by calibrating this same couple to a number of fixed temperatures, we are able to determine the temperature of the test.

The relation between the temperature and the electro-motive force may be expressed mathematically by the parabolic formula of two terms:

$$E = a + b\,(T - t) + b\,(T^2 - t^2),$$

in which

$T =$ the temperature of the hot junction,

$t =$ the temperature of the cold junction.

while a, b and c are three known measured temperatures such as the freezing points of zinc, antimony and copper.

This is an empirical formula originated by Averarius and Tait.

The two metals composing the element (as the thermo-electric couple is called) may be of great variety, but for an accurate couple we are very much limited in our choice of metals.

As the range of the apparatus should be as high as possible, platinum, fusing at 3,236 degrees Fahr., suggest itself as most desirable, while the extended researches of Le Chatelier showed that the best metal to be used in connection with platinum was an alloy of platinum with 10 per cent. of rhodium, which alloy has but a slightly lower fusing temperature than the platinum itself.

Experiment has taught that for an accurate and reliable couple the metal composing the element must be the purest obtainable, as well as homogeneous throughout, otherwise parasite currents are set up which are fatal to accuracy.

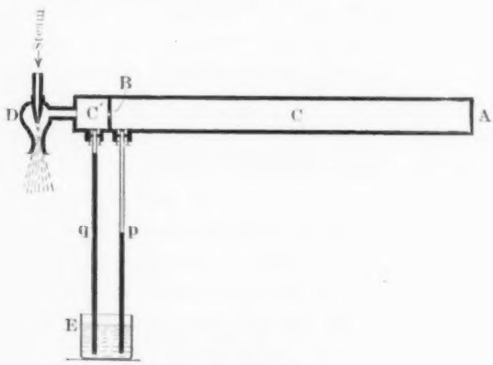


FIG. 11.

A good test for homogeniety is to connect the wires of a couple in a circuit and then heat them at different parts with the flame of a Bunsen burner. If the wires are homogeneous, the electromotive force set up in the circuit will be very small.

Iron, nickel, palladium, and their alloys are utterly unsuited for the measurement of high temperatures on account of the relatively intense parasite currents set up within them.

The metal used in elements should not oxidize readily at the temperatures at which they are used, and they should not be readily affected by the atmosphere and its impurities into which mixture they are plunged.

Platinum is, unfortunately, easily affected by many metallic vapors as well as carbon vapors, and silicon at high temperatures, so it is necessary to protect these elements carefully when used in such surroundings.

Many metals become brittle after long intermittent heating, doubtless due to crystalization, which is apt to affect their usefulness for accurate temperature measurements.

When platinum couples become thus brittle they are easily restored by heating them for several hours to a white heat, and this is best done by passing a current of electricity through them.

For the past seventy years many experimenters have used almost every known variety of metals to construct thermo-couples, but for many reasons, including those I have just named, none has proved as satisfactory and lasting as the platinum and platinum with 10 per cent. rhodium elements.

Professor Bristol, the most recent investigator in this field, has, to my knowledge, been working diligently a long time for a commercial solution of this problem, and for the successful application of cheaper metals than platinum, for elements.

Nearly all of the previous investigators have been working for the highest refinement in accuracy, and have left the cost of the apparatus they develop entirely out of consideration, as well as the delicacy of the instrument, which makes it almost a laboratory apparatus.

Professor Bristol, as I understand, has worked for the production of an apparatus which would be comparatively low in cost, and without going into ultra-refinements he has endeavored to produce a thermo-electric couple that would be sufficiently correct for all ordinary commercial and manufacturing applications.

In using thermo-couples, it is usually necessary to protect them from surrounding gases and vapors by some kind of an envelope. This is also necessary when they are plunged into molten metals. The protecting cover may be a steel pipe, but this cannot be used at temperatures exceeding 1,475 degrees Fahr. to 1,650 degrees Fahr., the tube becoming soft at the latter temperature.

I have used soft gray cast-iron pipes up to temperatures of 2,100 degrees Fahr., but when such pipe fails in the furnace it is apt to fly all to pieces rather than soften and droop. Asbestos is frequently used to cover the wires, but this will melt between 2,192 degrees Fahr. and 2,372 degrees Fahr.

For higher temperatures porcelain tubes are used, but they are very fragile and expensive, and when applied in a furnace care must be taken, first, to heat them slowly up to the furnace temperature, and, next, you must see that they are uniformly heated throughout their whole length.

These tubes can be used in temperatures running from 2,750 degrees Fahr. up to 2,900 degrees Fahr. Such tubes must be of a superior quality, and the best are manufactured by the Royal Berlin Porcelain Works.

When tubes surrounding an element are placed in a horizontal position they sometimes soften and bend. Damage to the tube and element can usually be avoided by slowly revolving the tube, which will cause bending in the opposite direction, and thus the tube may be straightened.

The two wires of the couple must be insulated one from the other when placed in such tubes, and this is accomplished by wrapping them with asbestos or slipping them in asbestos tubes, or by threading them through small fireclay pipes, such as made by Hall & Bro., of Buffalo, or else by using small porcelain tubes, which are made especially for this purpose.

The readings of temperatures taken when using thermo-electric couples show us the difference in temperature between the two ends of the couple.

By this I mean the temperature at the point where the two ends of the two metals are joined (known as the hot juncture) and the outer ends of these two wires, projecting outside of the furnace (known as the cold junction), where the copper leads connect them to the electrical measuring instrument.

Most couples are calibrated with their gold juncture immersed in ice water—*i.e.*, at 32 degrees Fahr.—and therefore, if a higher temperature exists during a test, at the cold juncture, corrections must be applied to the readings indicated.

This feature of the instrument sometimes leads to a serious error, especially when flame occasionally darts out of the opening through which the element is introduced.

In such cases I have usually drawn the cold juncture a little further out of the furnace so as to bend the wires down into a bath of ice water, but in ordinary cases I loop the two wires near the cold juncture so they will hold a thermometer and take readings from both thermometer and millivolt meter at the same time.

Professor Bristol has a very ingenious method for correcting this error, due to heated cold junctures, which must be generally appreciated. The correction of this error corresponds somewhat to the stem correction needed for mercurial thermometers, mentioned above, but no further correction of this kind is needed with the thermo-electric pyrometer, as it makes no difference whether

the whole element is immersed in the hot bath or whether the extreme end alone, including the hot juncture, is heated, the radius will be the same.

The thermo-electric pyrometer, like all other scientific and engineering apparatus, has its limitations. It can only be used to measure temperatures at short distances from the interior of a furnace wall, being thus limited by the length of the element as well as the length of its surrounding cover. It cannot be used to measure temperatures above the fusing point of the metal composing the element, nor the fusing or softening point of its surrounding protective casing.

When temperatures are required beyond these ranges of distance or temperature, I am using optical pyrometers. I have found the Warner optical pyrometer wonderfully accurate, and it can be as carefully calibrated from known points of fixed temperature as the thermo-electrical pyrometer, and besides, being constructed upon a purely mathematical basis, the instrument can be accurately graduated to read high degrees of temperature far beyond points where we have any means of calibration, and I have read temperatures in electric furnaces as high as 4,500 degrees Fahr., knowing that my error is not over 11 degrees.

Mr. E. A. Uehling.—The more we learn about the effect of temperature in operations in which heat is a controlling factor, the more we feel the want to get out of the realm of guessing, into that of knowing, and as means have been gradually developed to determine with a closer degree of approximation of the temperatures employed, and the more their influnces have been studied the more pronounced has become the desire for accurate knowledge of the heat requirements, and hence heat measurement.

From what we have seen here to-night and heard from Professor Bristol, I am sure that we are all satisfied that a most valuable addition has been made to the means for measuring accurately, moderately high temperatures.

I happen to be one of those of whom our President has said, that when entering from the college into the field of metallurgy, found a most deplorable lack of knowledge of temperatures employed and no means at all, or at best very inadequate means for measuring such temperatures. Although this was less true of the blast furnace operations, with which the writer became most intimately connected, yet the pyrometers in use were very unsatisfactory. From the time of the introduction of the hot blast by Neil-

son some attempt has been made to measure its temperatures. Bars of lead and zinc were used, which gave some indication from which a guess at the temperature could be made. Later the expansion pyrometer came into quite general use.

As the temperature of the blast was increased more and more, it became more and more difficult to measure the same. Recourse was then had to diluting the temperature of the blast by the introduction of cold air before passing over either the expansion rods or impinging upon the bulb of a mercury thermometer, as is the case respectively in the Brown and Hobson pyrometers.

Arriving at the temperature with these instruments is on a par with the method employed by the Irishman to get at the weight of his pig. He lays a plank across a log so that it extends equal distances on either side, then places the pig on one end of the plank and piles stones on the other end, until they balance accurately, and then guesses at the weight of the stones, which must necessarily be the weight of the pig. Now this method is not as absurd as it may appear at first glance. A man familiar with stones can guess at their weight much closer than he could at the weight of a hog.

Similarly, the instruments in use aided materially in guessing the temperature of the blast.

Feeling very keenly the want of an instrument which would measure temperatures accurately, the writer devised numerous more or less (generally less) satisfactory means for measuring high temperatures, none of which were enough better than those existing to be worthy of further development.

Having the matter continually on his mind, he finally struck upon the principle upon which the pneumatic pyrometer is based, which is as follows:

The pneumatic pyrometer is based on the laws governing the flow of air through small apertures.

If two such apertures, *A* and *B*, Fig. 11, respectively, form the inlet and outlet openings of a chamber *C*, and a uniform suction is created in the chamber *C′* by the aspirator *D*, the action will be as follows:

Air will be drawn through the aperture *B* into the chamber *C′*, creating suction in chamber *C*, which in turn causes air from the atmosphere to flow in through the aperture *A*. The velocity with which the air enters through *A* depends on the suction in the chamber *C*, and the velocity at which it flows out through *B*

depends upon the excess of suction in C' over that existing in the chamber C, that is, the effective suction in C'. The total suction remaining constant, the effective suction must decrease as the suction in C increases, hence the velocity at which air flows in through the aperture A increases and the velocity at which the air flows out through the aperture B decreases, until the same quantity of air enters at A as passes out at B. As soon as this occurs no further change of suction can take place in the chamber C.

If the apertures A and B are of the same size, and the temperature of the air is the same while flowing through both, then equilibrium is established when the suction in C is one-half the suction in C'.

Air is very materially expanded by heat. Therefore the higher the temperature of the air the greater the volume, and the smaller will be the quantity of air drawn through a given aperture by the same suction. Now if the air, as it passes through the aperture A is heated, but again cooled to a lower fixed temperature before it passes through the aperture B, less air will enter through the aperture A than is drawn out through the aperture B. Hence the suction in C must increase and the effective suction in C' must decrease, and in consequence the velocity of the air through A will increase and the velocity of the air through B will decrease, until the same quantity of air again flows through both apertures. Thus every change of temperature in the air entering through the aperture A will cause a corresponding change of suction in the Chamber C. If two monometer tubes p and g, Fig. 11, communicate respectively with the chambers C and C', the column in tube q will indicate the constant suction in C' and the column in tube p will indicate the variable suction in the chamber C, which suction is a true measure of the temperature of the air entering through the aperture A.

To embody the described principle in a practical pyrometer, the following conditions must be fulfilled:

a. The air must be drawn through the apertures with a constant and perfectly uniform suction.

b. The aperture A must be so disposed that it can be located in such a manner that the air, before passing through it, must have acquired the temperature which is to be indicated. The parts exposed to the heat must be constructed of material that will resist the highest heat to be determined.

c. The aperture B must be located in a medium of constant temperature.

d. Provision must be made that the apertures remain perfectly clean.

e. The chamber C must be absolutely tight, so that no air can enter except through the aperture A.

The complete instrument in the form used principally for annealing furnaces and similar work is illustrated in Fig. 12. The

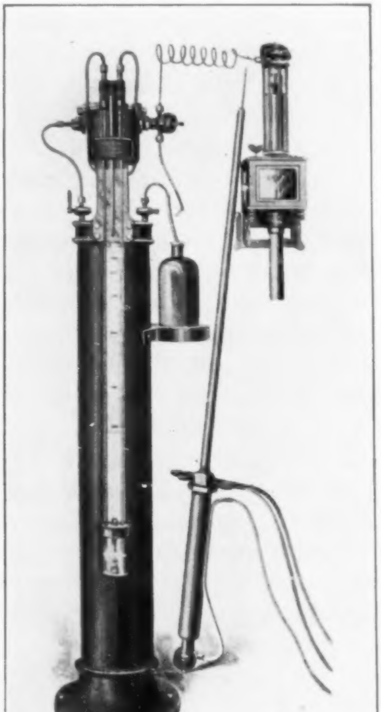


FIG. 12.

Standard Blast Furnace Pyrometer is shown in Fig. 13. It would take too much time to fully describe the pneumatic pyrometer here, but the writer will gladly send full description and give any other information desired to any one interested.

We have to-day three forms of pyrometers, each fulfilling the conditions of accurate heat measurement, based on three distinct

principles, first, the pneumatic; second, the electric, and third, the optic principle, each of which has its own peculiar advantages.

For temperatures above the melting point of platinum the optic pyrometer reigns supreme; for temperatures below 3,000 degrees Fahr., where readings are to be taken at many points in more or less rapid succession, the electric pyrometer has no rival. Both

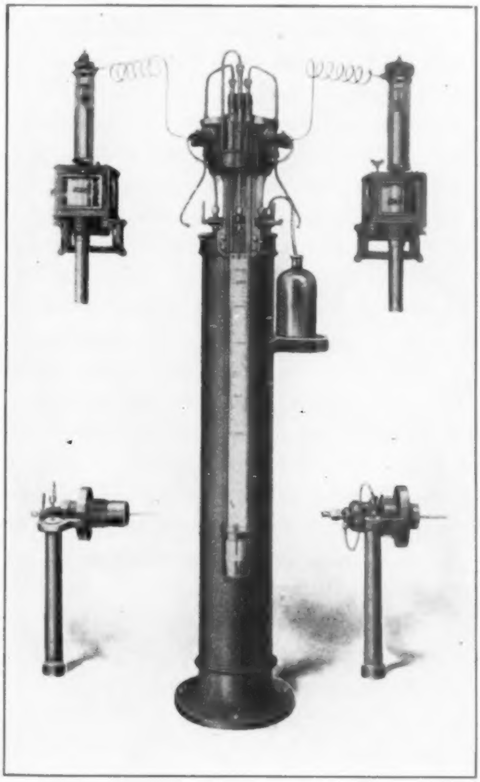


FIG. 13.

these instruments have the advantage of easy probability. Where temperatures are to be continuously indicated and recorded at the same place, or for longer or shorter intervals of time at places within a radius of moderate distance, say 100 to 200 feet, the pneumatic pyrometer answers better than either of the other two.

I consider the continuous autographic record of the temperature

involved in any process in which heat is a controlling factor of paramount importance.

In this respect the pneumatic pyrometer has no equal. Another feature in which the pneumatic pyrometer stands alone is that in addition to making a continuous autographic record it will, if found necessary or desirable, indicate the temperature at several places at the same time.

The pneumatic pyrometer has now been on the market for over ten years, and has become the standard blast furnace instrument. It is installed at over 75 per cent. of the blast furnaces in this country for indicating and recording both the temperatures of the blast and the top gas. It is extensively used in England and also on the Continent; three of them went to Japan.

The highest temperature measured with the pneumatic pyrometer was 3,350 degrees Fahr. This temperature was found in one of the tube-welding furnaces of the National Tube Works at McKeesport. For the first few hours it looked as though it might stand up under this high temperature; it responded promptly to the regulation of the fire, the opening of the furnace door, etc., but after about twelve hours it became evident that it indicated too low, and after thirty-six hours the pyrometer showed 2,000 degrees to 2,300 degrees, while the furnace temperature was practically the same as when the pyrometer was first inserted. On examination it was found that the inner and outer platinum tubes had become welded together. The metal had become so soft that the aperture had become enlarged by the current of air passing through it, and this caused the indication to be too low. The upper limit of temperature that can be measured with the pneumatic pyrometer to-day is the point where platinum begins to get very soft. For continuous indication 2,500, possibly 2,700 degrees Fahr. cannot safely be exceeded. For short time exposure 3,000 degrees can be measured, but for all heat measurements where an autographic record has no value, the electric pyrometer is to be preferred, and for occasional observations of temperatures above 2,000 degrees Fahr. Wahner's optical pyrometer is the one to use.

As already stated, an autographic record of the temperatures on which the pyrometer is to keep you posted is second in importance only to the accurate indication of the heat to be controlled. Fig. 14 A shows a full-sized twenty-four hour record of the pneumatic pyrometer of the temperature above the bridge wall of a steam boiler. This record shows how often the boiler was coaled,

when the fires were cleaned and the time consumed for this purpose. It also shows that the firing was done quite differently between 6 A.M. and 4 P.M. than from that time on. The maximum temperature of the fire was reached much quicker during former than that of the latter period, but the latter period shows much higher top heats. Without the autographic records these and other facts can be brought out only by very frequent reading and laborious tabulation, and at best only in a much less perfect and satisfactory way. For practical control of important heat operations the autographic record necessarily adds incalculably to the value of the pyrometer. The same principle upon which the pneumatic pyrometer is based has been utilized by the writer in the development of an instrument called Gas-Composimeter, which will continuously indicate and record the per cent. of CO_2 in any given gas, and hopes to bring this instrument before the Society as soon as he feels fully warranted to do so.

Fig. 14 B shows an autographic record of the CO_2 contained in the chimney gas of a steam boiler made by one of these instruments. It was simultaneously taken with the temperature record discussed above.

It will be noticed how similar these two records are, as they necessarily must be. It should be observed, however, that the change in the condition of the fire which caused the higher temperature over the bridgewall after 4 P.M. produced a much more pronounced increase in the CO_2 of the products of combustion, hence the inference seems warranted that the per cent. of CO_2 is a better indicator of economy of boiler firing than the temperature over the bridge wall.

Professor Bristol has acquainted us with a number of advantages possessed by his thermocouple over the platinum rhodium couple, of which the most important, to my mind, is its cheapness. This factor will accelerate the diffusion of knowledge of heat treatment and accentuate the necessity of accurate heat measurement, and thus broaden the field of practical application of all good pyrometers. There is nothing that has done more to retard the science of pyrometry than the use of cheap, inaccurate and unreliable pyrometers. A cheap, accurate and reliable instrument as that of Prof. Bristol promises to be and will have a very great stimulating effect.

*Prof. Wm. H. Bristol.**—For the rapid measurement of the

* Added after presentation of paper at monthly reunion.

temperatures of objects and metallic bodies the thermo-electric couple may be employed to great advantage. For illustration, if it is desired to obtain almost instantaneous indications of the temperature of a metallic plate, the ends of the elements forming the hot end of the couple, may be left disconnected and reduced to points at their extremities. If the pointed ends of such a couple are pressed against the surface of the plate at the desired point, the metal will serve as the electric conductor between the end points of the elements, and their junctions will immediately assume the temperature of the plate, giving a corresponding instantaneous response on the indicating instrument.

By simply pressing the points of the couple at different places on the metallic body the different temperature at the various points may quickly and accurately be determined.

For application of this form of the couple to the measurement of the temperature of objects that are not electric conductors, a thin metallic sheet may be placed on the surface of the object and the couple then pressed against it.

To insure that the temperature of the junction is that of the object itself, a sheet or layer of heat resisting material, as asbestos, may be applied to the object at the point when the temperature is desired.

The elements of the couple may be joined in the usual manner and then pressed against the object. When so made this method of measuring the temperature of solid objects is equally applicable, whether the objects are electric conductors or not.

To make the low-resistance thermo-electric pyrometer applicable for measurement of temperatures as high as 3,000 degrees Fahr. where the couple is to be exposed to the full heat to be measured, metals like those of the Le Chatelier couples or their equivalents having high fusing points must be employed. A compound couple, which will serve the same purpose as that of platinum-rhodium, has been devised, but which is less expensive.

It consists of two parts, which, together, form the complete couple, as indicated in Fig. 15. The part which is exposed to the full heat to be measured is made of platinum-rhodium and is of sufficient length to reach a point where the temperature will not exceed 1,200 degrees Fahr. From this point to the extreme cold ends of the couple the elements are made of inexpensive alloys, as indicated in Fig. 15. This portion of the couple is of ample cross-section to eliminate changes of resistance that would otherwise be

produced by variations of temperature along the length of the couple.

Two thermo-electric junctions, B and C, are introduced into the circuit where connection is made between the low-priced alloys and the platinum-rhodium elements, but the electro-motive forces generated at these points may be made equal and opposed to each other if the proper alloys are employed.

Although no common metals or alloys have as yet been found which give a perfect balance or neutralization of the thermo-electric effects for wide variations of temperature at the secondary junctions B and C, it has been determined that alloys of iron and nickel will give a very close balance for ranges of temperature from the atmosphere to 1,200 degrees Fahr. at these points. For this range of increase of temperature at the secondary junctions there

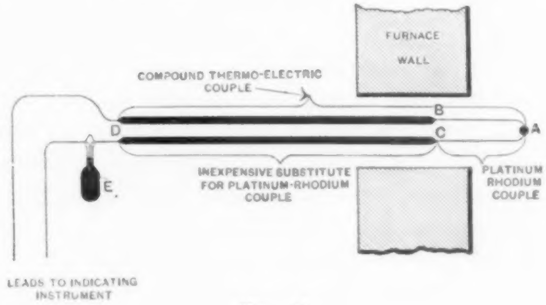


FIG. 15.

is a small positive excess of electro-motive force which may be compensated for by the introduction of an auxiliary resistance into the circuit in close proximity to the secondary junction, the resistance being placed in series and made of material which will increase in resistance with the rise of temperature at the junction, and consequently the compound couple taken as a whole will be the same as if the entire couple was made of the platinum-rhodium elements.

Automatic continuous records of the indications of this pyrometer may readily be made on a chart sheet which is arranged to move at the proper rate just behind the end of the indicating arm, but not in the least interfering with its natural motion. The record sheet is unsupported over its active portion and is periodically vibrated by the clock movement into contact with the end of the indicating arm and produces a record upon the chart sheet

showing its position at the instant of vibrations. By timing the period between the vibrations of the chart the contacts may be made so as to produce a continuous record.

The record may be made by ink carried by the indicating arm or the surface of the record sheet may be coated with some easily removable substance.

For recording automatically rapid changes of temperature, a current from an induction coil may be passed from the end of the indicating arm through the chart at frequent intervals.

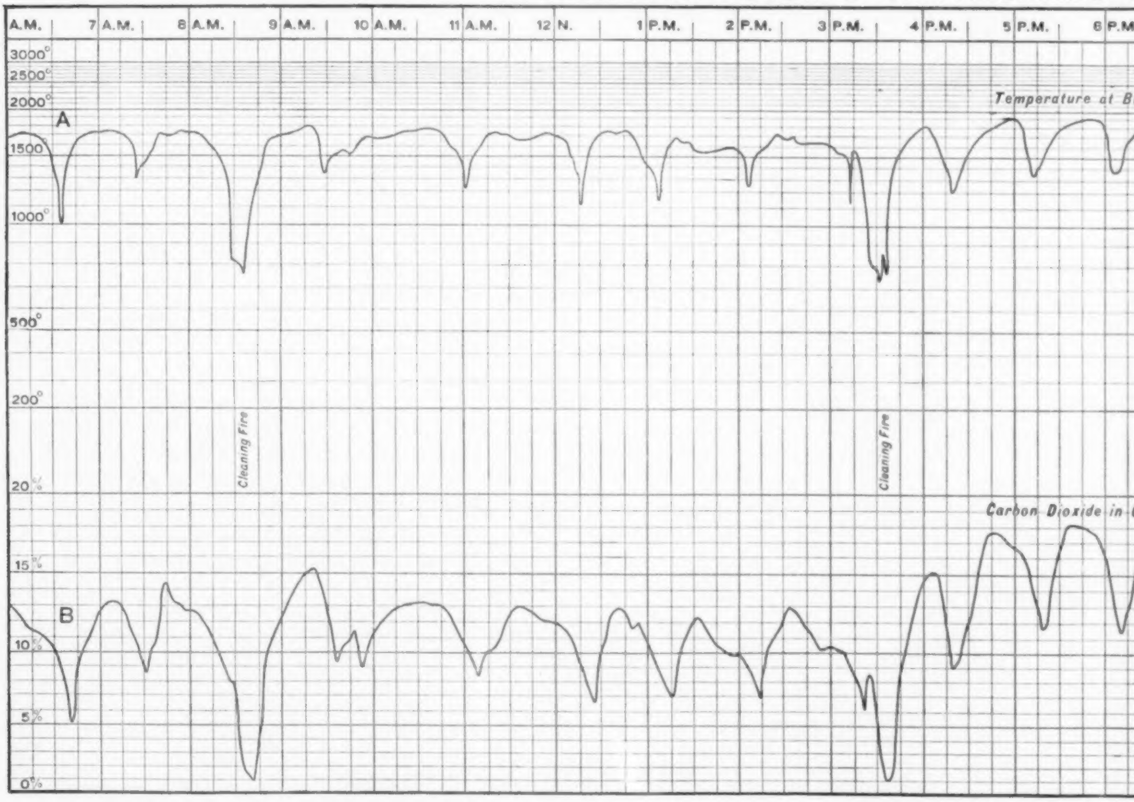
Mr. George H. Barrus.—I would like to ask Professor Bristol what the need of a compensator is on his instrument, when none is used on the Le Chatelier?

Mr. Bristol.—The Le Chatelier needs one and needs it badly, for with this instrument it is necessary to make corrections for the changes of temperature at the cold end of the couple in order to obtain accurate results. A thermo-electric couple depends for its operation upon the difference of temperature between the two ends; and whatever metals or alloys are used the electromotive force varies in some proportion to the difference in temperature between the hot and the cold ends. Therefore, to obtain perfect results, the temperature at the cold end must be maintained constant or readings must be taken with some other thermometer at the cold end and corrections made accordingly. The compensator device obviates the necessity of either providing the constant temperature at the cold end or of taking the readings and making the corrections. It is automatic in its operation.

I have known of a case where the Le Chatelier couple was used in connection with an annealing or malleable iron furnace where no provision was made for keeping the temperature of the cold end at a uniform temperature and where it was so hot that the man in charge of it had to use heavy gloves in handling it.

I calibrate these instruments, as I have described in the paper, by the use of the melting points of different metals, and, in conjunction with that, I also use a standard Le Chatelier pyrometer which has been calibrated by the Reichsanstalt Laboratory at Berlin, where the recognized standards of temperature are maintained. To calibrate by the melting points of metals I generally take copper, lead and zinc, using a molten bath. The couple is immersed into the molten metal, and kept there until it chills. There is a very decided length of time at the freezing point of the metal at which the temperature remains constant and there

AUTOGRAPHIC RECORDS OF PNEUMATIC P



Above shows fac-simile of two 24-hour autographic records made respectively by the Pneumatic Pyrometer and the Gas Composimet carbon-di-oxide contained in the chimney gas.

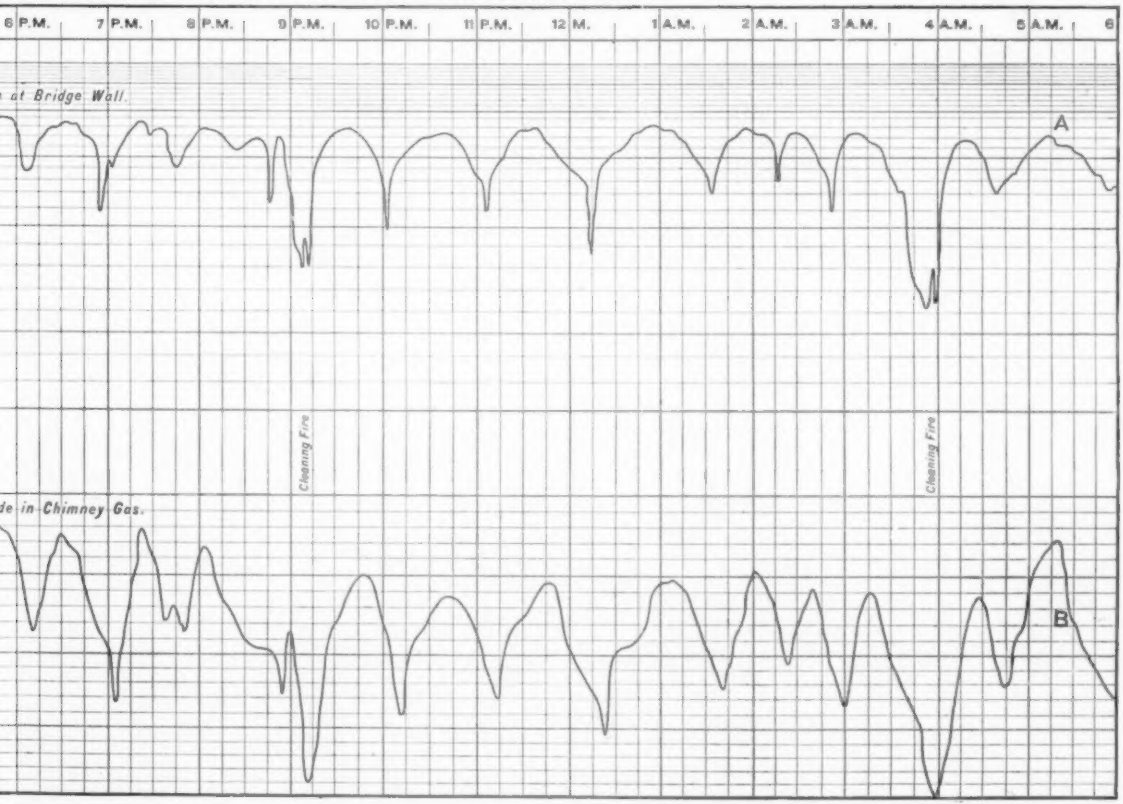
The sudden drops in temperature are caused by opening the fire door for the purpose of coaling up. The record, therefore, gives the ex

When the fire door is opened the cold air rushing in reduces the temperature, and, since it at the same time dilutes the products of c perfect in that respect, which accordance makes each a proof of the accuracy of the other.

FIG. 14.

Prof. Wm. H. Bristol.

C PYROMETER AND GAS COMPOSIMETER.



osimeter from a steam boiler. A A is a record of the temperature above the bridge wall. B B is a simultaneous record of the per cent. of

the exact time and number of coalings. It further shows that the fires were cleaned at 3:45, 8:30 A. M., and also at 3:30 and 9 P. M.
ts of combustion, it is evident that the per cent. of CO_2 must also drop, and the accordance of the two records A A and B B could not be more

14.

is no possibility of making a mistake in your reading of that temperature. In a mass of metal of a few pounds the temperature may be held constant for half a minute or more at the freezing point. It is like ice water, where the temperature will stand at 32 degrees until the ice is all melted, except in freezing the action is just the reverse. As molten metal in a crucible freezes and the center is still in a molten condition, the temperature will remain practically constant until it is all frozen.

Mr. A. Bement.—Keiser & Schmidt of Germany make a form of D'Arsonval galvanometer, wherein the moving coil is not supported by a suspending wire. I have used this instrument with the high resistance elements and have never known of a case of damage to it; while on the other hand, those forms using the suspension by a strand require handling with extreme care.

President Taylor.—I have used the Le Chatelier pyrometer very extensively, and I can say that it is a most unsatisfactory instrument, as everyone knows who has tried to use it, unless all of the surrounding conditions are favorable. If you are using it in the ordinary shop in which furnaces are used for manufacturing articles commercially, the Le Chatelier pyrometer is an extremely difficult instrument to use. In our case we immersed the cold end in a stream of cold water in order to keep the temperature of the end uniform, as without doing so it would have been impossible to obtain satisfactory results, as we had to take so many readings and continue them for such a length of time that some apparatus of that sort was necessary.

I want to say for all of us who have used pyrometers that I think Mr. Bristol's invention, if it accomplishes all that is claimed for it, is invaluable. It meets a demand which has existed during the last twenty-five years and I am ashamed to say that many of us during that period have wasted a large amount of time in attempting to do, without success, what he has apparently accomplished.

*Prof. Wm. H. Bristol.**—In closing this paper I wish to express my appreciation of the complimentary and encouraging remarks made by President Taylor and others who have taken part in the discussion.

I trust that the instrument in continued and practical everyday service will fulfill all expectations.

* Author's closure, under the Rules.

No. 1107.*

EFFICIENCY TESTS OF TURBINE WATER WHEELS.†

BY WM. O. WEBBER, BOSTON, MASS.
(Member of the Society.)

Abstract.

These are records of a series of consecutive tests under uniform conditions of four water wheels, with the same apparatus and the same observers, to determine the efficiency of the wheels, and ascertain the effects of corrosion and wear upon the relative efficiency of the wheels, also to determine whether more modern wheels would show greater relative efficiencies, and whether it would be more economical to operate all of the wheels at part gate all of the time, or some of the wheels at full gate part of the time.

The tests are instructive because of the wide range of speeds and gate openings covered systematically and consecutively. They show conclusively that water wheels should be run at the speed for which they are designed under any given head, that modern wheels, with coarser buckets having a spoon-shaped discharge depending below the guide ring of the wheel, and with the outer periphery of the bucket at discharge of greater diameter than the bucket at entrance, would give much more power, higher speed, and greater efficiencies from the same water than from wheels of older design without these modern improvements.

They also show how rapidly a complete series of efficiency tests

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* Presented at the Chattanooga meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

† For further discussion on this topic, consult *Transactions* as follows :

No. 61, vol. 3, p. 84 : "Efficiency of Turbines, as affected by Form of Gate." Wm. O. Webber.

No. 243, vol. 8, p. 359 : "Systematic Testing of Turbine Water Wheels." R. H. Thurston.

No. 665, vol. 17, p. 41 : "Water Power : Its Generation and Transmission." Samuel Webber.

No. 666, vol. 17, p. 58 : "Water Power : Caratunk Falls, Kennebec River, Maine." Samuel McElroy.

of a wheel can be made with properly designed apparatus efficiently handled by a corps of intelligent observers.

1. These tests were made to determine the efficiency of several turbine water wheels, of the mixed inward and downward flow type, which has gradually been developed from the inward flow reaction wheel, sometimes called the Francis turbine. Some of them had been running for a considerable period of time, and it was desirable, if possible, to determine what effect, if any, corrosion, wear, and other processes of deterioration had had upon the relative efficiency of the wheels, as compared with the efficiency of the same wheels when new, and the relative efficiencies of wheels of a similar character, as constructed at the present time, and also to determine the relative efficiencies of the wheels at diferent speeds and different gate openings.

2. The wheels consisted of two vertical, Risdon wheels, 54 inches in diameter, made by the Holyoke Machine Company, about thirty years ago, and were installed in separate pen stocks, side by side with crown gears engaging with beveled gears on a horizontal jack shaft. They were cylinder gate wheels, with rather closely pitched buckets, the buckets being of the same diameter at the discharge as at the entrance, and at full gate were full reaction wheels; at half gate, or less, they were simply action wheels. Each wheel vented 350 inches of water, and, under 24 feet head, would discharge 95 cubic feet a second, and give 221 horse-power at 128 revolutions a minute, equal to 70 per cent. of the spouting velocity of the water, under which conditions the wheels should give their best efficiencies.

3. These wheels were called respectively the North and South Risdon Wheels, and were used to furnish the power for two Worthington, double-acting, power pumps of 17½ by 38-inch stroke plungers. The North Risdon wheel buckets were all whole, both above and below the bottom of the flume. In the South Risdon wheel the buckets were O. K. below the flume bottom, but three of the buckets had pieces, about 6 inches long and 2 inches deep, broken out of them above the flume bottom. This wheel was corroded and covered with barnacles, which had to be scraped off before gate could be fully opened.

4. Another wheel, installed in another flume, was a 40-inch Risdon, double capacity wheel, and was used to drive an electric light dynamo, in the same manner, through crown gear meshing into a beveled gear upon a horizontal shaft. This wheel was of a

later development than the other Risdon wheels, having enlarged buckets and a larger diameter at the discharge than at the entrance of the wheel bucket, and, although normally a 40-inch wheel, it vented practically the same amount of water as the 54-inch, that is, 340 inches of water, with a discharge of 93 cubic feet a second, giving 215 horse-power at 173 revolutions per minute. This wheel was designated as the Electric Light Wheel, and was about ten or twelve years old. It was in perfect order

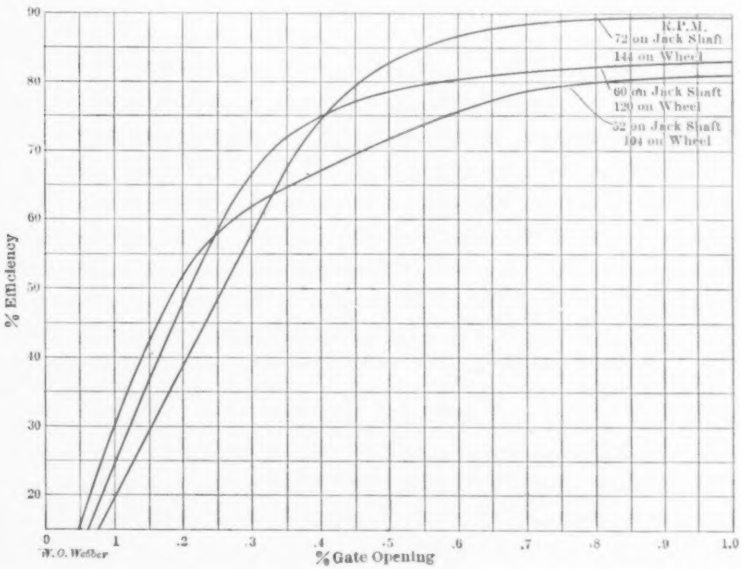

FIG. 1.—EFFICIENCY CURVES. 39-INCH HERCULES WHEEL.

below the flume bottom, but had three small nicks in the buckets above the flume bottom.

5. The fourth wheel was a 39-inch Hercules wheel, geared, as before, to a Deane geared, double-acting pump, $16\frac{1}{2}$ by $35\frac{1}{2}$-inch plungers, the ratio of the crown gear on wheel to beveled gear on jack shaft being 2 to 1. This wheel was of a later development, in which the pitch of the buckets was much coarser, the depth of the buckets much greater in proportion to the diameter, and with the bottom of the buckets depending below the guide ring of the wheel, and of a spoon shape, so as to have both downward, inward and outward flow at discharge. The outer periphery of the bucket at discharge was of larger diameter than the bucket at

entrance. It vented 466 inches, delivered 127 cubic feet of water per second, and gave 276 horse-power at 146 revolutions per minute, all on the basis of 24 feet head.

6. It is not known what the maker's guarantees were on the Risdon wheels at the time they were originally installed, but probably from 70 to 75 per cent. The Hercules wheel was probably guaranteed for very nearly 80 per cent.

7. The jack shafts were uncoupled from the pumps, so that

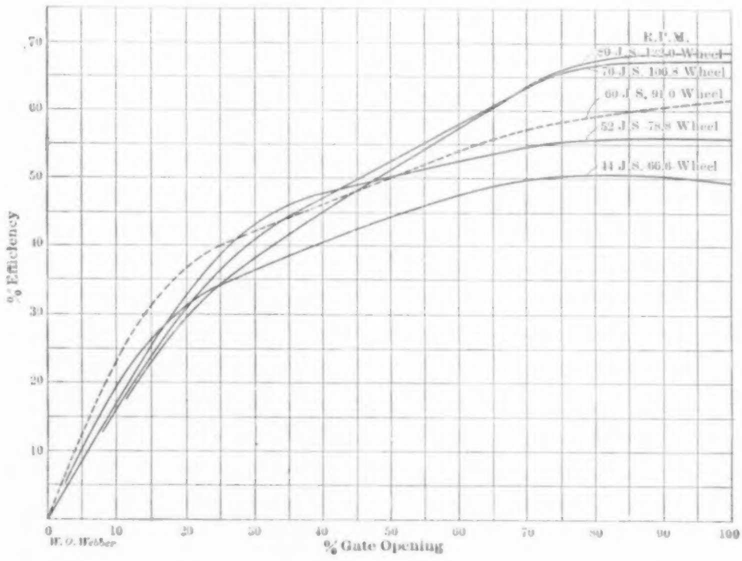

FIG. 2.—EFFICIENCY CURVES. SOUTH NO. 1. 54-INCH RISDON WHEEL.

each wheel could be tested separately, and the pony brake was applied directly to the shaft in a horizontal position. The brake apparatus consisted of a 48-inch diameter by 26¼-inch face brake pulley rim, fastened by slotted holes to two spiders fitted to the jack shaft. Into the end of this shaft was screwed a stud, carrying a worm on its outer end, engaging a worm wheel having 100 teeth, operating a gong which struck once in every 100 revolutions of the wheel shaft. The brake rim had an outside flange to hold the brake in place, and an interior flange to hold water inside the rim of the wheel.

8. The brake clamp consisted of 4½ x ⅜-inch steel bands, hinged together and fitted with 3-inch square by 26-inch long maple

blocks, entirely enveloping the band wheel. They were clamped together by two $1\frac{3}{4}$-inch diameter screws passing through two clamp castings attached to the ends of the brake bands. These screws were operated by a horizontal hand-wheel shaft, having a 30-inch hand wheel with exterior spokes, and a ratio of gears of two to one to the clamp screws. The under side of the gears fitted to the clamp screws, and the wrought iron nuts into which

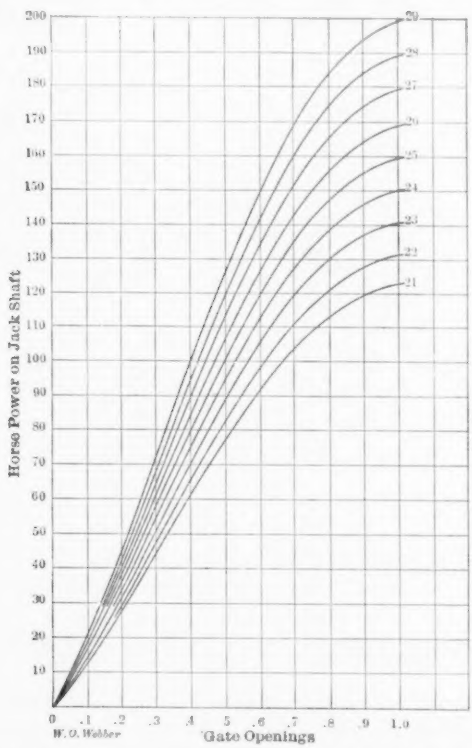

FIG. 3.—POWER CURVES. SOUTH NO. 1. 54-INCH RISDON WHEEL ON WORTHINGTON PUMPS. JACK SHAFT 50 R. P. M. WHEEL 75.7 R. P. M.

the screws were entered formed ball and socket joints with the clamp castings above mentioned.

9. The main brake beam consisted of a 6-inch, $14\frac{3}{4}$-pound I-beam, securely held in the upper of the two clamp castings. At a point 10 feet 6 inches from the center of the brake, on the under side of this I-beam, was located a V-block, the sides of the V being at an angle of 90 degrees. Situated directly above

this point, but extending at right angles thereto, was a wrought iron lever of 10 to 1 leverage; the shorter arm being 6 inches and the longer arm being 60 inches. This lever was fitted with three 1½-inch knife edges. From the shorter end of this lever a link connected with a clevis sustained the outer end of the main brake beam.

10. The middle fulcrum was carried on two steel plates hav-

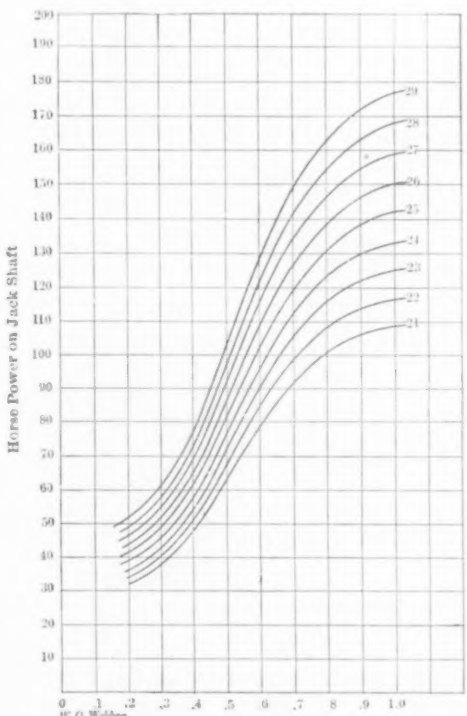

FIG. 4.—POWER CURVES. SOUTH NO. 1. 54-INCH RISDON WHEEL ON WORTHINGTON PUMPS. JACK SHAFT 43 R. P. M. WHEEL 65 R. P. M.

ing half-round grooves planed in them, and situated on the top of two oak posts securely bolted to an adjoining brick wall. To these same posts were bolted two blocks confining the vertical motion of the main brake beam to a distance of 3 inches. From the longer end of this lever depended a scale pan, provided with weights varying from 5 to 20 pounds each. Depending from the main brake beam, at a point about 8 feet from the center of the

brake, was a hinged rod and piston loosely fitted into an 18-inch dash pot filled with crude oil.

11. Into the interior of the brake wheel were laid two streams of cold water, under city pressure, through $\frac{3}{4}$-inch pipes, and drawing the warm water from the interior of the brake wheel were two 1-inch siphon scoop pipes, 1 inch in diameter. The brake was lubricated with beef tallow, forced into the front and rear openings of the brake by hand and by wooden paddles.

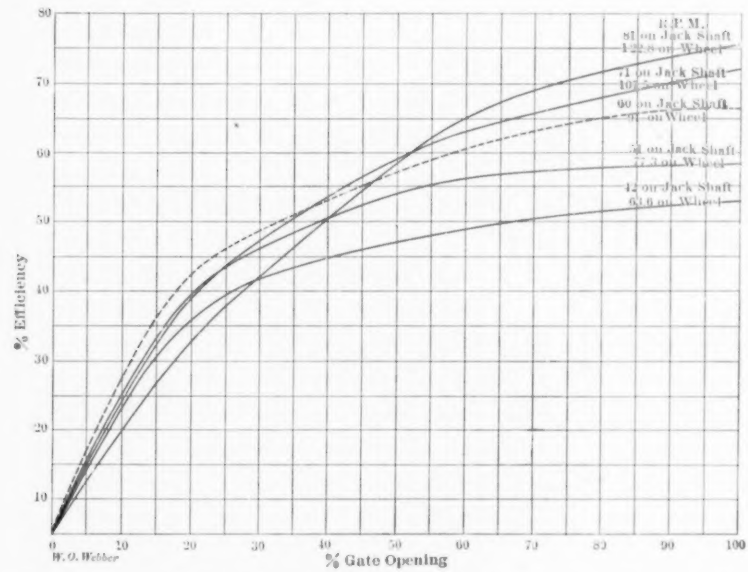

FIG. 5.—EFFICIENCY CURVES. NORTH NO. 2. 54-INCH RISDON WHEEL.

12. A float gauge was established in the flume leading to each wheel and within 10 feet of the wheel in each instance. This float gauge was carefully set and leveled from datum points established by the Water Power Company. A weir was constructed below the wheel house, into which all of the tail-races emptied. The crest of the weir was 4 feet above the bottom of the tail-races, and the weir opening was 29.43 feet long. A perforated pipe was submerged directly back of the weir, and a hook gauge was set up directly connecting with a measuring can, which was, in turn, connected to the perforated pipe. This hook gauge was read at one minute intervals. A recording hook gauge was also set up, and connected in similar manner, and the readings checked.

The crest of the weir and both hook gauges were carefully leveled and verified by two persons from the Water Power Company's datum.

13. The leakage on the Risdon wheels was first determined, and one-half of the leakage was subtracted from the water required for the North Risdon wheel. The leakage of all the

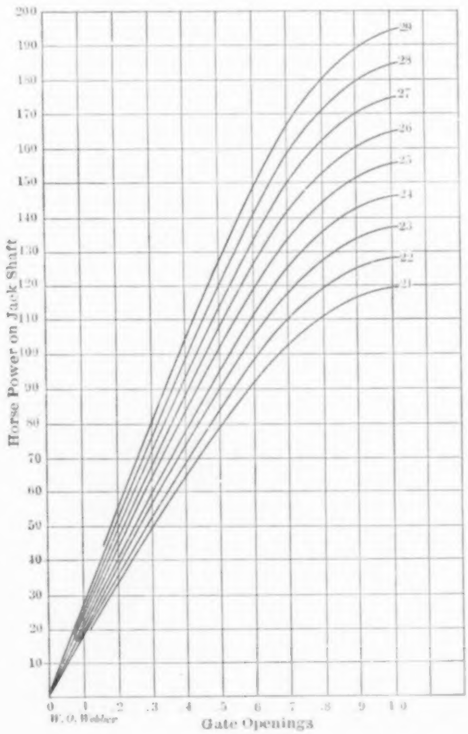

FIG. 6.—POWER CURVES. NORTH NO. 2. 54-INCH RISDON WHEEL ON WORTHINGTON PUMPS. JACK SHAFT 50 R. P. M. WHEEL 75.7 R. P. M.

wheels was then measured, and three-fourths of this leakage was deducted from the South Risdon wheel. A similar deduction was made in reference to the 39-inch Hercules wheel. The leakage of the three Risdon wheels was then determined, and two-thirds of this leakage deducted from the Electric Light wheel. The leakage of all the wheels was then determined, and another

column prepared showing the efficiencies of all the wheels with all the leakage deducted.

14. The first tests were made on the 39-inch Hercules wheel, and consisted of a full gate test, with the jack shaft running at 60 revolutions per minute, corresponding to a speed of 120 revolutions per minute on the wheel. This test was continued for ten minutes until all of the readings became steady. A test at approximately three-fourths gate opening was then made of the

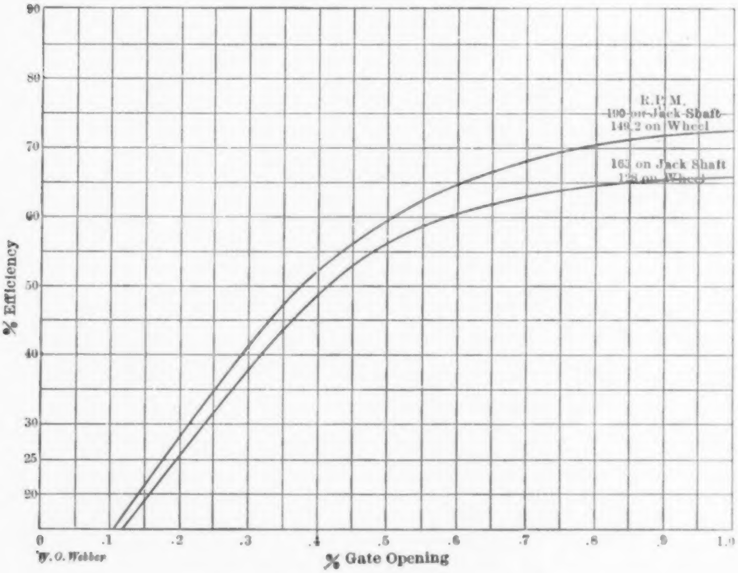

FIG. 7.—EFFICIENCY CURVES. ELECTRIC LIGHT WHEEL. 40-INCH RISDON DOUBLE CAP.

same wheel, at the same speed, then at half gate, then at three-tenths gate, and another at about one-sixth gate.

15. These were succeeded by tests at approximately 140 revolutions per minute of the wheel at three-fourths, half, and a quarter gate, and these again by tests with similar gate openings, at 104 revolutions per minute of the wheel. These tests showed that the best results on this wheel were obtained at a speed of 144 revolutions, which is very nearly the proper speed for this sized wheel, under 23 feet head. The crossing of the efficiency lines at different speeds demonstrates how important it is that water wheels should be run at the proper speed for which they

were designed, under a given head (see Fig. 1). This wheel, at the speed of 144 revolutions per minute, arrived at its maximum efficiencies at about 0.85 gate opening, and continued at the same rate of efficiency to full gate opening.

16. At 150 revolutions per minute this wheel would probably show a still higher efficiency at full gate opening, but would show a lower efficiency at half gate opening than when running at 144. The test is also remarkable in showing the high efficiencies of 75 per cent. at 0.4 gate, 85 per cent. at half gate, and 88 per cent. at three-quarter gate, and would undoubtedly show over

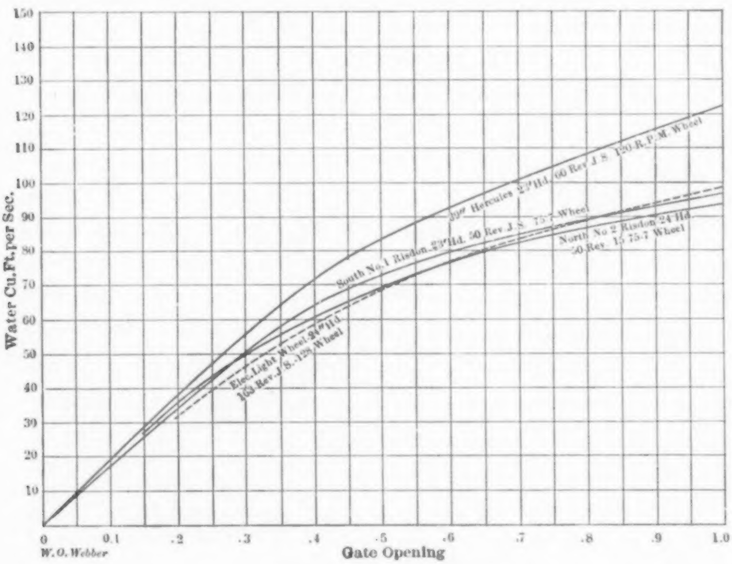

Fig. 8.—Water Curves.

90 per cent. efficiency at full gate under the best conditions of speed.

17. The next series of tests on the 54-inch South Risdon wheel were made at quarter, three-eighths, one-half, three-quarter, and full gate openings, with the wheel running at 67, 79, 91, 107 and 122 revolutions per minute. This wheel was the one with the broken buckets, and the results are shown by the flat places in the curves on Fig. 2. This flatness in the curves, however, is a characteristic of all tests of Risdon wheels.

18. Power curves for this wheel are shown in Fig 3, under different heads, at 75.7 revolutions per minute, which is a speed lower

than the wheel should be run, hence the reverse in the curves. A second diagram, Fig. 4, at a still lower speed, shows a greater reverse in the curves, and the effect of the loss of power owing to water squirting through the broken buckets, at a velocity greater than the velocity of the wheel. In other words, when the velocity of the wheel corresponds nearly to the velocity of the water, under a given head, the effect of leakage is not so apparent.

19. The next series of tests were on the 54-inch North Risdon wheel, at the same variety of gate openings, and, as nearly as possible, the same speeds. This wheel was in better shape, had

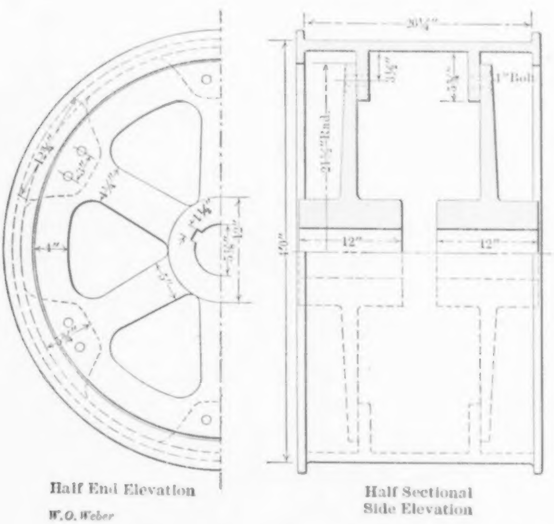

FIG. 9.—BRAKE WHEEL OF PRONY BRAKE FOR TESTING 6-8 H.-P. TURBINE.

no broken buckets, and gave a much better efficiency, and is also freer from the flatness in the curves. It shows again, in a most marked manner, the effect of speed on efficiency (Fig. 5).

20. Fig. 6 shows the power curves of this wheel, in which the reverse of the curvature is noticeably absent.

21. The last series of tests were on the 40-inch Risdon Electric Light Wheel, at 0.4, 0.5, 0.6, 0.75, and full gate, respectively, at 128 and 149 revolutions per minute on the wheel, which are both below the proper speed for this wheel, under the conditions obtained (see curves, Fig. 7).

22. Fig. 8 shows the amount of water, in cubic feet per

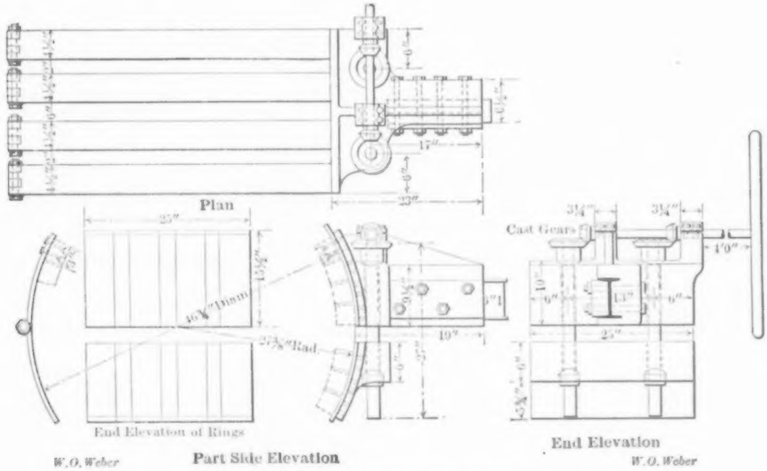

FIG. 10.—DETAIL OF PRONY BRAKE BANDS.

second, used by each of the wheels, at different percentages of gate opening, and at the different heads under which the tests were made. The reason that the curve of the South Risdon wheel is higher than that of the North Risdon wheel is undoubtedly due to the leakage of that wheel. The difference between

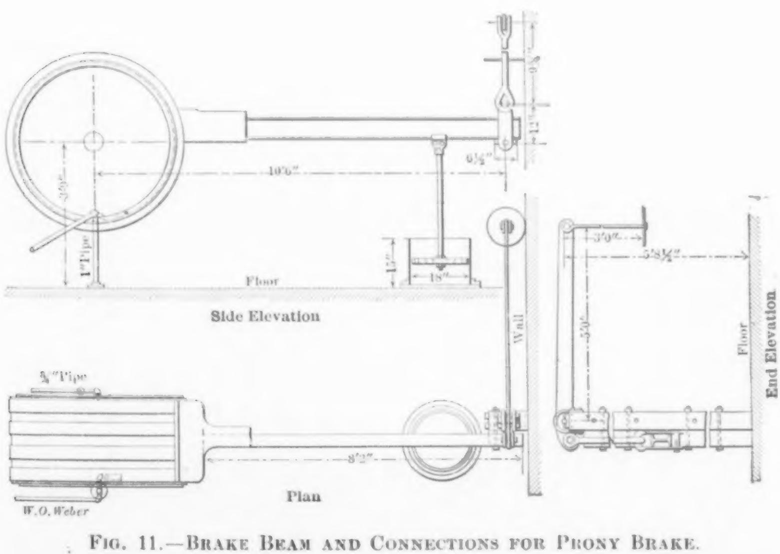

FIG. 11.—BRAKE BEAM AND CONNECTIONS FOR PRONY BRAKE.

the Risdon wheels and the Hercules wheel is due to the difference in design of the two wheels, and shows a much larger ventage of water in the 39-inch Hercules wheel over the 40-inch Electric Light wheel, and an increasing ratio of ventage from half gate upward.

23. These tests are particularly instructive and interesting, because of the wide range of speeds and gate openings covered systematically and consecutively, under practically uniform conditions, and with the same apparatus and the same observers.

54-INCH NORTH RISDON WHEELS.

APRIL 22, 1905.

Ratio of Jack Shaft to Wheels 1 : 1.513.

Date and Time.	No. of Test.	Per Cent. of Gate Opening.	Rev. Per Min. of Jack Shaft to which Brake was Applied.	Weight on Prony Brake 10 feet Radius.	Brake Horse-power of Wheels, Gears, Bearings and Shafts.	Net Head of Water on Wheel.	Cu. Ft. Sec. for Weir 29.43 feet long.	Water Horse-power of all Water Passing Wheels.	Brake Horse-power of Wheels Less 5% for Gears, etc.	Net Efficiency of Wheels.	Net Efficiency of Wheels Deducting ‡ Leakage.
1	2	3	4	5	6	7	8	9	10	11	12
Apr. 22.									(6 + 5%)	(10 ÷ 9)	
9.30–38	1	.25	40.9	60.000	46.73	24.711	51.5937	144.69	49.06	33.93	38.56
9.42–50	2	"	51.66	55.000	54.10	24.716	52.1882	146.30	56.80	38.84	44.06
9.55–10	3	"	59.125	50.000	56.29	24.730	52.3088	146.69	59.10	40.29	45.70
10.33–08	4	"	73.25	40.000	55.79	24.7 9	52.4266	147.04	58.58	39.84	45.18
10.10–15	5	"	82.00	30.000	46.84	24.742	52.0704	148.07	49.18	33.22	38.21
10.18–23	6	.375	81.00	50.000	77.11	24.529	64.9961	179.02	80.97	45.23	48.35
10.25–30	7	"	70.80	60.000	80.88	24.527	65.7024	182.76	84.93	46.47	51.31
10.32–37	8	"	62.30	70.000	83.03	24.533	66.1527	184.06	87.18	47.37	52.26
10.39–45	9	"	52.13	80.000	79.41	24.514	67.3152	187.11	83.37	44.56	49.06
10.47–52	10	"	44.00	90.000	75.40	24.519	67.9655	188.99	79.17	41.89	46.09
10.55–11	11	.50	49.50	100.000	94.25	24.371	75.7175	209.30	98.96	47.28	51.50
11.00–07	12	"	42.60	105.000	85.16	24.385	75.8528	209.72	89.42	42.64	46.42
11.12–17	13	"	62.00	90.000	106.25	24.399	75.7851	209.64	111.56	53.22	59.32
11.19–25	14	"	69.50	80.000	105.86	24.433	75.4585	209.09	111.16	53.16	57.92
11.26–31	15	"	79.70	70.000	106.24	24.474	75.9882	210.90	111.55	52.89	57.59
11.40–43	16	.75	81.60	105.000	163.14	24.076	94.6409	258.50	171.30	66.27	70.93
11.45–50	17	"	73.90	115.000	161.82	24.075	95.5121	260.72	169.91	65.17	69.69
11.53–59	18	"	58.10	130.000	143.81	24.201	94.6409	259.79	151.00	58.12	62.20
12.03–08	19	"	50.80	135.000	130.58	24.258	93.1900	256.38	137.01	53.44	57.25
12.09–14	20	"	43.00	145.000	118.74	24.285	92.5867	254.97	124.68	48.90	52.40
12.17–22	21	1.00	41.20	165.000	129.43	24.147	103 3405	283.03	135.90	48.02	51.08
12.23–27	22	"	52.00	155.000	153.46	24.157	104.5412	286.53	161.13	56.24	59.67
12.29–34	23	"	60.00	150.000	171.36	24.177	104.8414	287.50	179.93	62.58	66.52
12.35–39	24	"	71.25	140.000	189.92	24.176	106.4983	292.10	199.42	68.27	72.51
12.40–45	25	"	80.25	130.000	198.64	24.224	107.4077	295.06	208.57	70.69	75.01

54-INCH SOUTH RISDON WHEELS.

APRIL 20, 1905.

Ratio of Jack Shaft to Wheels 1 : 1.513.

Date and Time.	No. of Test.	Per Cent. of Gate Opening.	Rev. Per Min. of Jack Shaft to which Brake was Applied.	Weight on Prony Brake 10 feet Radius.	Brake Horse-power of Wheels, Gears, Bearings and Shafts.	Net Head of Water on Wheel.	Cu. Ft. Sec. for Weir 29.43 feet long.	Water Horse-power of all Water passing Wheels.	Brake Horse-power of Wheels Less 5% for Gears, etc.	Net Efficiency of Wheels.	Net Efficiency of Wheels Deducting ? Leakage.
1	2	3	4	5	6	7	8	9	10	11	12
April 20									(6 + 5%)	(10 ÷ 9)	
8.24–34	1	.250	43.14	50.000	41.07	23.829	55.44	149.82	43.12	28.77	33.97
8.38–45	2	.375	43.50	65.000	53.83	23.691	62.797	169.13	56.52	33.42	38.72
8.50–9.04	3	.500	43.54	90.000	74.61	23.462	75.205	199.14	78.34	39.34	44.12
9.25–31	4	.750	44.40	135.000	114.13	23.013	98.340	256.60	119.84	46.70	51.10
9.35–43	5	1.000	45.50	150.000	129.95	22.720	111.834	288.17	136.45	47.35	51.24
9.47–52	6	1.000	54.00	145.000	149.08	22.704	112.846	290.49	156.53	53.88	58.25
9.55–59	7	.750	53.40	130.000	132.18	22.934	101.848	264.89	138.79	52.39	57.15
10.03–07	8	.500	51.80	95.000	93.57	23.290	82.922	219.16	98.24	44.84	49.96
10.10–15	9	.375	49.75	75.000	71.04	23.510	70.228	187.13	74.59	39.94	45.30
10.23–27	10	.250	51.43	45.000	44.07	23.778	53.097	135.25	46.27	34.21	38.44
10.20–34	11	.250	57.50	40.000	43.79	23.790	52.726	142.26	45.98	32.31	38.51
10.36–40	12	.375	60.00	55.000	62.89	23.640	62.018	158.94	66.03	39.09	45.15
10.44–50	13	.500	62.00	75.000	88.54	23.410	76.176	202.22	92.96	45.87	51.72
10.52–58	14	.750	62.00	115.000	135.75	22.980	98.932	257.85	142.22	55.16	60.33
11.00–05	15	1.000	59.00	140.000	157.27	22.670	113.835	292.68	165.13	56.42	60.96
11.21–27	16	1.000	69.50	130.000	172.02	22.636	113.293	290.86	180.62	62.10	67.12
11.29–35	17	.750	69.00	115.000	151.08	22.916	102.446	266.30	158.63	59.57	65.00
11.38–41	18	.500	72.00	70.000	95.96	23.376	79.046	209.61	100.76	48.07	53.85
11.43–47	19	.375	68.75	50.000	65.45	23.637	64.422	172.72	68.72	39.69	45.82
11.50–56	20	.250	70.25	30.000	40.13	23.948	50.767	137.89	42.14	30.56	36.69
11.57-12.02	21	.250	76.50	25.000	36.41	24.190	50.531	138.63	38.23	27.58	33.13
12.7–10	22	.375	79.75	40.000	60.74	24.032	62.765	171.08	63.78	37.28	43.11
12.13–17	23	.600	80.50	55.000	84.30	23.893	75.503	205.53	88.51	43.27	48.73
12.19–24	24	.750	80.50	100.000	153.27	23.525	99.179	264.56	160.93	60.82	66.50
12.25–29	25	1.000	81.00	120.000	185.07	23.271	114.300	301.67	194.32	64.41	69.58

39-INCH HERCULES WHEEL.

APRIL 15, 1905.

Ratio of Jack Shaft to Wheels 1 : 2.

Date and Time.	No. of Test.	Per Cent. of Gate Opening.	Rev. Per Min. of Jack Shaft to which Brake was Applied.	Weight on Prony Brake 10 feet Radius.	Brake Horse-power of Wheels, Gears, Bearings and Shaft.	Net Head of Water on Wheel.	Cu. Ft. Sec. for Weir 29.43 feet Long.	Water Horse-power of all Water Passing Wheels.	Brake Horse-power of Wheels Less 5% for Gears, etc.	Net Efficiency of Wheels.	Net Efficiency of Wheels Deducting ¾ Leakage.
1	2	3	4	5	6	7	8	9	10	11	12
April 15									(6 + 5%)	(10 ÷ 9)	
1.29–32	1	1.00	60.00	240.000	274.18	23.178	131.899	346.64	287.88	83.05	88.74
1.34–39	2	.76	60.00	205.000	234.11	23.936	116.316	315.69	245.86	77.88	83.99
2.08–13	3	.525	58.75	160.000	178.89	24.217	94.64	259.87	187.83	72.28	79.37
2.16–21	4	.291	64.44	75.000	91.92	24.803	61.126	171.90	96.52	56.14	65.17
2.27–35	5	.146	64.00	25.000	30.46	25.798	39.865	116.61	31.98	27.42	34.82
2.39–46	6	.437	71.60	110.000	149.95	24.522	81.038	225.32	157.45	69.88	77.97
3.05–10	7	.742	70.00	170.000	226.57	23.815	110.30	297.85	237.90	79.87	86.51
3.14–23	8	.291	73.25	60.000	83.68	24.886	63.78	179.97	87.86	48.82	56.65
3.24–28	9	.291	52.40	90.000	89.79	24.860	62.638	176.56	94.28	53.96	61.75
3.29–34	10	.437	53.00	140.000	141.28	24.438	84.934	235.35	148.34	63.03	70.00
3.51–57	11	.742	51.80	220.000	216.98	23.662	116.007	311.24	227.83	73.20	78.95

40-INCH RISDON ELECTRIC LIGHT WHEEL.

APRIL 26, 1905.

Ratio of Jack Shaft to Wheels 1 : 1.272.

Date and Time.	No. of Test.	Per Cent of. Gate Opening.	Rev. Per Min. of Jack Shaft to which Brake was Applied.	Weight on Prony Brake 10 feet Radius.	Brake Horse-power of Wheels, Gears, Bearings and Shaft.	Net Head of Water on Wheel.	Cu. Ft. Sec. for Weir 29.43 feet Long.	Water Horse-power of all Water Passing Wheels.	Brake Horse-power of Wheels Less 5% for Gears, etc.	Net Efficiency of Wheels.	Net Efficiency of Wheels Deducting ½ Leakage.
1	2	3	4	5	6	7	8	9	10	11	12
									(6 + 5%)	(10 ÷ 9)	
2.50–5	1	.4	163.8	25.000	78.00	24.606	62.930	175.62	81.93	46.76	48.94
2.59–3.04	2	.5	163.8	32.800	103.14	24.374	42.755	199.55	108.30	54.27	56.59
3.13–17	3	.6	160.	38.840	118.32	24.276	79.001	217.50	124.24	57.12	59.34
3.22–26	4	.75	163.2	46.340	144.05	24.024	89.799	244.66	151.25	61.82	63.93
3.31–35	5	1.00	162.9	53.840	167.14	23.760	101.701	274.05	175.50	64.04	65.96
3.51–55	6	.4	188.	22.500	80.7	24.626	61.679	172.26	84.74	49.19	51.67
3.57–4.02	7	.5	192.5	28.840	105.81	24.332	69.940	193.00	111.10	57.56	60.11
4.05–09	8	.6	190.1	35.000	126.93	24.269	77.886	214.38	133.28	60.75	64.61
4.12–16	9	.75	187.5	42.500	151.75	24.023	88.431	240.94	159.34	66.13	68.43
4.19–22	10	1.00	183.4	50.000	174.68	23.759	99.961	269.36	183.41	68.09	70.18

No. 1108.*

EFFECT OF A BLOW.

BY ALEXANDER W. MOSELEY, CHICAGO, ILL.
(Member of the Society.)
AND JOHN LORD BACON, CHICAGO, ILL.
(Junior Member of the Society.)

1. Some months ago one of the writers standardized some copper plugs to be used to measure the pressure exerted by an embossing press. At that time it occurred to him that similar methods might be applied to a much greater extent than at present to measure the effects of machines and especially to measure the effects of blows.

2. Power and steam hammers are ordinarily rated by the weight of the moving parts. The weight and shape of the anvil, foundation, frame, anvil block and hammer block, in fact, the design of the hammer in almost every detail as well as the weight and shape of the hammer, rod and piston, the steam pressure on both sides of the piston, and the amount of drop, enter into the effect of the blow.

3. The effect of a blow is the measure of what a blow does to the piece struck; it is expressed in units of work, and the change in form of the piece struck can be made to show the number of foot pounds absorbed by it.

4. Fig. 1 gives the plot of the permanent lengths given to two plugs cut from a coppered Bessemer steel rod. Each was placed in an Olsen testing machine, loaded to the amount indicated, removed and calipered in length, replaced in the machine, loaded to the next higher amount, removed and calipered, and so on to the largest loads indicated. The plugs were nominally $\frac{1}{2}$-inch diameter by $\frac{1}{2}$-inch long, but they departed from this somewhat. One was 0.5177-inch longer by 0.500-inch diameter and the points for it are indicated on Fig. 1 by circles; the other was 0.5145-inch long by 0.498-inch diameter, indicated by crosses. The ordinates in Fig. 1 are expressed in decimals of the original length.

5. Curve A, Fig. 2, shows the plot between work and length.

* Presented at the Chattanooga Meeting (May, 1906) of the American Society of Mechanical Engineers, and forming a part of Volume 27 of the *Transactions*.

The work corresponding to each length was found by planimetering the areas—viz., in Fig. 1, the area between the axis of lengths, the curve, and the horizontal through 0.80 represents 145.4 foot pounds. These are plotted in curve A, Fig. 2.

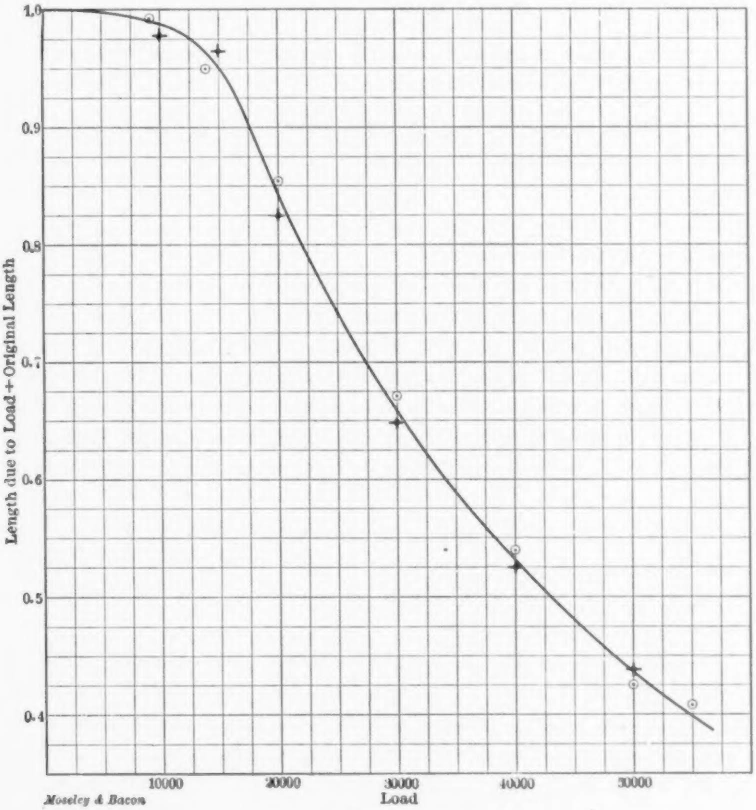

FIG. 1.

Curve A, Fig. 2, is the standard work curve for the $\frac{1}{2}$-inch by $\frac{1}{2}$-inch plug.

6. Curve B, Fig. 2, shows the effects of the blows of a Bement-Miles 200-pound steam hammer under known amounts of kinetic energy. The hammer, piston and piston-rod weigh 195 pounds and the hammer can be raised to a height of 13.5 inches. Rods of lengths 6, 8, 10, 12.1 and 13.4 inches long, respectively, were used to support the hammer above the anvil. After carefully locating the plug on the center of the anvil, the support was knocked

from under the hammer. In measuring the kinetic energy the length of the plug after the blow was subtracted from the length of the rod. (The cylinder was removed and the packing loosened during these tests.)

7. The following details are recorded:

All plugs were annealed by heating to redness and allowing to cool in the air.

All plugs were of Bessemer steel rod.

In the testing machine the plugs were oiled with a thin film of light oil and compressed between hardened and ground steel plates.

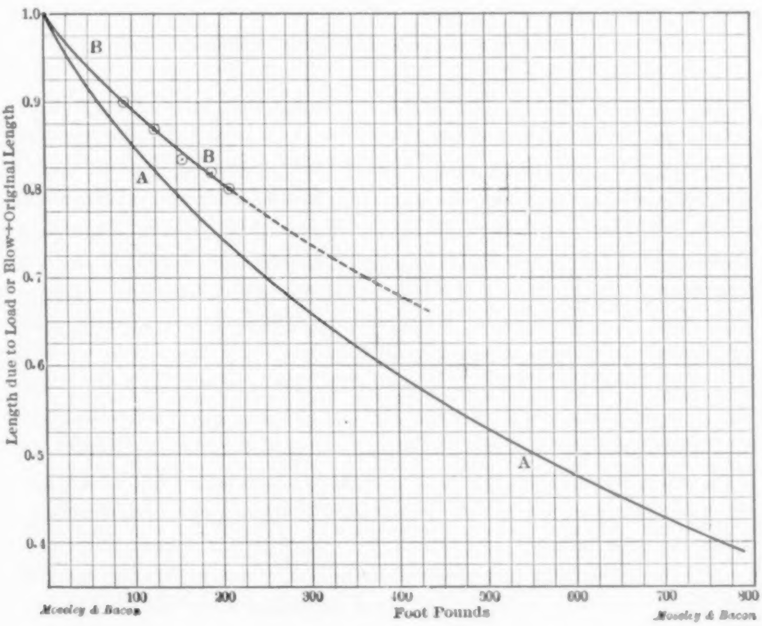

FIG. 2.

In the drop tests (without steam) the plugs were oiled.

The anvil block had been newly dressed and was slightly rough from the tool.

The rebound of the hammer seemed negligible and no attempt was made to allow for it.

8. The annealing is a matter of importance, as is evident from the following, obtained in the testing machine on $\frac{1}{2}$-inch by $\frac{1}{2}$-inch plugs:

Change in length from

unit length to	0.90	0.80	0.70	0.60	0.50	0.40
Foot pounds annealed	60.8	145.4	251.7	383.1	550.3	760.1
Foot pounds unannealed	66.2	156.4	262.9	404.3	570.9	787.1.

9. On comparing the curve of the effects of the blows with the curve of work, it is apparent that a part of the energy is lost; in this particular case, between lengths 1.00 and 0.80, it requires about 1.45 units of kinetic energy to do 1.00 unit of work on the plug. This means a loss of effect in the blow of about 31 per cent.

10. Drop tests and work tests were made on $\frac{3}{8}$-inch diameter by $\frac{3}{8}$-inch plugs between the lengths 1.00 and 0.60 with a resulting loss of about 29 per cent.

11. Owing to the fact that the largest kinetic energy due to the drop of the hammer under gravity was only sufficient to change the length of the plugs from 1.00 to 0.80, curve B, Fig. 2, may not be reliable for reductions in length by a blow to less than about 0.75. To test the full effect under steam of the hammer mentioned above, at first three and then four plugs were put on the anvil together. In the former trial the aggregate effect of blow was 489 foot pounds (taken from curve A), and the kinetic energy was 692 foot pounds (from curve B). In the latter case the aggregate effect of the blow was 616 foot pounds and the kinetic energy was 872 foot pounds. Both blows were intended to be the heaviest blows of the hammer. It was decided to consider the difference in the strength of the blows as due to "accidental" causes. A reasonably certain "heaviest blow" could best be ascertained after several trials.

12. Each of two $\frac{1}{2}$-inch by $\frac{1}{2}$-inch plugs was given the full blow of the hammer. Curve A showed that one absorbed 675 and the other 705 foot pounds with a mean for the two of 690 foot pounds. In view of the marked difference cited in paragraph 11, it seems fair to assume that each case is a measure of the actual energy absorbed from the blow, and that the total (mean) kinetic energy was about 690 x 1.45 or 1,000 foot pounds, of which 310 foot pounds were lost.

13. By the methods suggested, two important results can be accomplished for any particular hammer, viz.:

a. With a standard plug the effect of a blow can be measured.

b. With a standard plug and the kinetic energy of the hammer at the instant of the blow known the loss of energy can be found.

Further, it would seem that such standard plugs made of ma-

terials and dimensions to suit the size of hammer and to insure the absorption of the hammer energy without injury to the hammer and anvil blocks, would serve as the means of rating hammers for purposes of comparison and sale.

All tests were made in the Strength of Materials Laboratory and Forge Shop of Lewis Institute.

DISCUSSION.

Prof. D. S. Jacobus.—I do not think the method adopted by the authors is logical, as the hammer struck the plugs at a much higher speed when operated with steam pressure than it did when the calibrations were made, and the hammer was simply allowed to fall on a plug. The correct way to attack this problem would be to investigate the effect of striking the plugs at different velocities with a falling weight so adjusted that the work done in falling would remain constant, and to note whether under these conditions the amount that the plugs are compressed is independent of the speed of the blow. That is, experiments should be made to show whether twice the weight falling one-half of the distance, or four times the weight falling one-quarter of the distance, will give the same compression as that found in the tests recorded in the paper. Unless the amount of compression is shown to be independent of the speed of the blow it is not logical to use one set of experiments, in which the mass of the falling weight is not varied, and assume that the ratio between the work required to compress the plugs a given amount with the falling weight and with a gradually increasing load will hold for any velocity at which the hammer may strike the plugs.

*Messrs. Moseley and Bacon.**—The thing sought was the measure of the effect of a blow. The work required to change the shape of the plugs was measured in the testing machine under pressure with velocity a negligible quantity. The writers have considered the likelihood of errors in the areas of Fig. 1 due to the elongation of the plugs after their removal from the machine. This error is small due to the small amount of recovery in length under such conditions. Curve A gives the work done on the plugs and hence the measure of the effect of the blow. The criticism is in reality a suggestion for further experiments somewhat aside from the immediate object of the paper. The results of such experiments would be valuable.

* Author's closure, under the Rules.

No. 1109.*

REPORT OF COMMITTEE APPOINTED TO COÖPERATE WITH THE PENNSYLVANIA RAILROAD SYSTEM IN CONDUCTING TESTS OF LOCOMOTIVES AT THE LOUISIANA PURCHASE EXPOSITION.†

To the American Society of Mechanical Engineers:

1. At the Saratoga meeting (June, 1903) the Society received a communication over the signature of Mr. J. J. Turner and Mr. Theodore N. Ely, representing the Pennsylvania Railroad System, and Mr. Willard A. Smith, representing the Louisiana Purchase Exposition, setting forth certain plans of the Pennsylvania System with reference to tests of locomotives at St. Louis, and requesting that a committee of the Society be appointed to cooperate with the officials of the railroad system in the advancement of this work. On motion, the Society voted that such a committee

* Presented at the Chattanooga Meeting (May, 1906) of the American Society of Mechanical Engineers and forming part of Volume 27 of the *Transactions.*

† In formulating its report, the committee of this Society has worked in conjunction with a similar committee of the Master Mechanics' Association, the effort being to secure a document which would serve both organizations as a joint report. As a consequence of this arrangement, a report similar in all essential respects with this will be presented by the committees of the Master Mechanics' Association to the Annual Convention of the Association in June.

† For further discussion on this topic consult *Transactions*, as follows:

No. 490, vol. xiii., p. 427: "An Experimental Locomotive." W. F. M. Goss.

No. 542, vol. xiv., p. 826: "Tests of the Locomotive at the Laboratory of Purdue University." W. F. M. Goss.

No. 552, vol. xiv., p. 1312: "Report of Committee on a Standard Method of Conducting Locomotive Tests."

No. 885, vol. xxii., p. 453: "Tests of the Boiler of the Purdue Locomotive." W. F. M. Goss.

No. 1032, vol. xxv., p. 550: "Road Tests of Consolidation Freight Locomotives." E. A. Hitchcock.

No. 1033, vol. xxv., p. 589: "Testing Locomotives in England."

No. 1039, vol. xxv., p. 827: "Locomotive Testing Plants." W. F. M. Goss.

be appointed, and the President subsequently named the undersigned to constitute its membership.

Your committee thus constituted would now report that it has performed the functions for which it was appointed, and presents herewith a statement covering the part which it has taken. It presents also a brief summary of the results which have been obtained from the tests.

2. *The Pennsylvania Organization.*—The responsibility for the tests at St. Louis rested with the following officials of the Pennsylvania System:

J. J. Turner, Third Vice-President, Pennsylvania Lines West of Pittsburgh.

Theodore N. Ely, Chief of Motive Power, Pennsylvania Railroad System.

F. D. Casanave, Special Agent, Pennsylvania Railroad System.

E. D. Nelson, Engineer of Tests, Pennsylvania Railroad System.

Coöperating with these were the heads of certain scientific and technical departments of the Railroad System, and especially Mr. A. W. Gibbs, General Superintendent Motive Power, and Mr. A. S. Vogt, Mechanical Engineer. The actual work at the plant was conducted by Mr. G. L. Wall, Director of Tests at St. Louis, assisted by an efficient staff of assistants, observers, computers and engine-men.

The part taken by the Pennsylvania System in proposing tests, in perfecting an organization for carrying them out, and in its execution of the various phases of the work, has been one of unusual significance.

It has brought into existence an entirely new testing plant, designed for mounting either freight or passenger locomotives, and capable of absorbing, for an indefinite period, the maximum power of a modern locomotive when running at any desired rate of speed;

It has caused to be designed and constructed a dynamometer capable of registering the tractive power of the heaviest locomotive, and at the same time so sensitive as to indicate the slightest variation in the force it may exert;

It has purchased and standardized instruments and apparatus for use in securing all data which has been deemed to be of scientific interest;

It has organized a complete corps of observers, engineers and

computers to carry out the tests, and to record, tabulate and analyze the results;

It has invited and secured the coöperation of scientific and technical men of this and other countries to assist in placing the tests upon the highest scientific plane possible in such work;

It has overcome difficulties, in many cases perplexing and serious, incident to the carrying out of such work as a part of a great International Exposition;

It has, as a result of its effort, defined the action of eight different typical locomotives, as regards the performance of the boiler, the engine, and of the locomotive as a whole, under many different conditions of operation, making of record a mass of information concerning the economic performance of the modern locomotive of great immediate value, and supplying a basis of comparison which will prove useful for many years to come;

It has met the expense of equipping and operating the plant with an unstinted hand, always holding considerations of cost to be subordinate to the definite object of making the tests as complete and valuable as possible, notwithstanding the fact that the amounts involved have been far greater than have ever been appropriated to any similar undertaking; and

It has carried out a broad plan of publication which has resulted in making data derived from tests, and all conclusions based thereon, together with a description of methods and means employed, all in great detail, accessible to the railroad officials and locomotive designers throughout the world.

3. *The Louisiana Purchase Exposition* was represented in the work by the Honorable Willard A. Smith, Chief of the Department of Transportation Exhibits.

4. *The Advisory Committee.*—At the time the invitation to coöperate was received by the American Society of Mechanical Engineers, a similar communication was sent to the American Railway Master Mechanics' Association, asking that organization to appoint a committee of three which would serve with a similar committee appointed by this Society. Individual invitations were extended also to certain distinguished engineers. The full membership of the Advisory Committee is as follows:

Representing the American Society of Mechanical Engineers:

W. F. M. Goss (Chairman), Dean of the Schools of Engineering, Purdue University.

Edwin M. Herr, General Manager, Westinghouse Air Brake Company.

J. E. Sague, First Vice-President, American Locomotive Company.

Representing the American Railway Master Mechanics' Association:

F. H. Clark, General Superintendent of Motive Power, Chicago, Burlington & Quincy Railroad.

* C. H. Quereau, Superintendent of Shops, New York Central & Hudson River Railroad.

H. H. Vaughan (Secretary), Superintendent of Motive Power, Canadian Pacific Railway.

Affiliated Members:

John A. F. Aspinall, General Manager, Lancaster & Yorkshire Railway, England.

Karl Steinbiss, Director, Royal Prussian Railway, Altona, Germany.

H. V. Wille, Assistant Superintendent, Baldwin Locomotive Works.

Throughout the progress of the work the committee served as an expert board of advisers. It assisted in formulating plans, it approved methods and reviewed results. Individual members of the committees, also, had an active part in giving shape to many matters of detail. The members have observed with unusual fidelity the appointments for the eight formal meetings of the committee, which have been as follows:

(1) 1903, June 25—Grand Union Hotel, Saratoga, N. Y.
(2) 1903, October 28—Union Station, Pittsburgh, Pa.
(3) 1904, January 28—Union Station, Pittsburgh, Pa.
(4) 1904, May 11—Transportation Building, St. Louis Fair.
(5) 1904, July 26—Transportation Building, St. Louis Fair.
(6) 1904, November 10—Transportation Building, St. Louis Fair.
(7) 1905, May 1—Broad Street Station, Philadelphia, Pa.
(8) 1905, June 12—Broad Street Station, Philadelphia, Pa.

5. *The Results* of the tests are well set forth in a formal publication of the Pennsylvania Railway System, entitled "Locomotive Tests and Exhibits, Pennsylvania Railway System, Louisiana

* During Mr. Quereau's absence from duty in the months of August, September, October and November, Mr. F. M. Whyte, Mechanical Engineer, N. Y. C. & H. R. R. R., was appointed by the Executive Committee of the American Railway Master Mechanics' Association to serve on the Advisory Committee.

Purchase Exposition," * a volume of 727 pages, containing approximately 900 illustrations.

In attempting to discuss results, your committee can only present brief abstracts of the published volume. These have, however, been selected and arranged to include those facts which are likely to be of interest to the members of the Society. The abstract thus arranged is submitted as an appendix to this report.

6. Your Committee, having completed the work for which it was appointed, and having hereby rendered its report to the Society, asks that it be now discharged.

Respectfully submitted,

W. F. M. Goss,
Edwin M. Herr,
J. E. Sague.

APPENDIX.

THE TESTS AND THEIR RESULTS.†

7. *Concerning General Conditions.*—The fuel used for all tests was a bituminous coal of high quality. The heating value of one pound of dry coal averaged more than 14,000 B. T. U. Its composition was as follows:

Fixed carbon	75.85
Volatile combustible	16.25
Ash	7.00
Moisture	.90

Great care was taken to have the coal fired as uniformly as possible, and to this end certain selected men served continuously in the capacity of firemen.

All locomotive tests were run in accord with a fixed schedule of speed. To avoid irregularities arising from differences in diameter of drivers, the speeds of this schedule were selected with reference to the revolutions of the drivers. The standard speeds

* It is the understanding of your committee that an edition of a thousand copies has been issued and that five hundred copies have already been distributed. The remaining number have been deposited with Mr. D. S. Newball, Purchasing Agent, Pennsylvania Railway Company, Broad Street Station, Philadelphia, from whom they may be obtained at a nominal price.

† For a description of the testing plant, see "Locomotive Testing Plants," Volume 25, page 827.

for freight locomotives were 40, 80, 120, 160 revolutions per minute, and for passenger locomotives 80, 120, 160, 240 and 320 revolutions. The preparations for observing the action of a locomotive during a test was elaborate, and all instruments occupied similar positions on all machines. The thoroughness which characterized the work may be judged from the fact that the log sheets for each test contained 399 items, many of which were the averaged values of many observations.

8. *The Locomotives Tested.*—Eight locomotives were tested, four having been designed for freight and four for passenger service. Two of the freight locomotives were simple and two were compound, while all of the passenger locomotives were of the four-cylinder balanced compound type, one being of French design and manufacture, one of German, and two of American. A summary of the locomotive tested is as follows:

THE LOCOMOTIVE TESTED.

Presented for Test by	Designating Number.	Service.	Wheel-Arrangement.	Description.	By whom manufactured.
P. R. R.	1,499	Freight.	2-8-0	Simple.	Pa. Railroad Co.
L. S. & M. S. Ry.	734	"	2-8-0	"	American Loco. Co.
Mich. Central	585	"	2-8-0	2-cylinder compound.	American Loco. Co.
A. T. & S. F. Ry.	929	"	2-10-2	4-cylinder tandem comp'nd.	Baldwin Loco. Works.
A. T. & S. F. Ry.	535	Passenger.	4-4-2	4-cylinder balanced comp'd.	Baldwin Loco. Works.
Hanover Loco. Works	628	"	4-4-2	4-cylinder balanced compound with superheater.	Hanover Loco. Works.
N. Y. C. & H. R. R. R.	3,000	"	4-4-2	4-cylinder balanced comp'd.	American Loco. Co.
P. R. R.	2,512	"	4-4-2	4-cylinder balanced comp'd.	Alsacienne Co.

THE LOCOMOTIVES TESTED AND RESULTS OBTAINED FROM THEM.

9. *Tests of Consolidation Locomotive, Pennsylvania Railroad Company.*—The first locomotive placed on the testing plant was No. 1,499, owned by the Pennsylvania Railroad Company. It is of the simplest consolidation (2-8-0) type, and is the standard heavy freight locomotive used on the Pennsylvania Railroad. It is known as the "H6a" type, according to the railroad company's classification. The locomotive was new and had not been thoroughly broken in before being tested. The first official test was made on May 25th, the locomotive having been run previously for three weeks in order to break in the plant. The prin-

cipal dimensions and details of the locomotive are shown in the following table:

Total weight, lbs	194,200
Weight on drivers, lbs	173,200
Cylinders (simple), inches	22 x 28
Diameter of drivers, inches	56
Fire-box heating surface, sq. ft	1,664
Heating surface in tubes (water side), sq. ft	2,667.27
Total heating surface (based on water side of tubes), sq. ft	2,843.67
* Total heating surface (based on fire side of tubes), sq. ft	2,482.26
Grate area, sq. ft	49.2
Boiler pressure, sq. ft	205
Valves	Richardson balanced
Link motion	Stephenson
Fire-box, type	Belpaire
No. of tubes	373
Outside diameter of tubes, inches	2
Length of tube, inches	164.5

* Used in calculations.

The maximum tractive effort was 39,773 pounds, which was calculated on the assumption that 80 per cent. of the boiler pres-

TABLE 1

SUMMARY OF DATA. PENNSYLVANIA RAILROAD LOCOMOTIVE NO. 1499.
Built at Juniata Shops of the Pennsylvania Railroad, Altoona, Pa., March, 1904.

	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
110	40.3	6.7	22.44	Full.	12,870	5.18	373	1,116	11.53	366	85	15,706	28.33	2.94	3.84
111	40.4	6.7	30.45	"	15,372	6.19	445	1,368	11.24	454	81	20,864	27.29	2.91	3.54
103	92.7	15.4	22.80	"	20,581	8.29	596	2,089	9.85	650	133	12,587	25.51	3.13	3.94
109	81.6	13.5	20.88	"	18,375	7.40	533	2,094	8.78	588	106	13,314	25.31	3.47	4.24
112	79.7	13.2	29.24	"	22,825	9.20	662	2,767	8.25	779	150	17,831	23.92	3.48	4.30
118	80.7	13.4	39.34	"	28,102	11.32	815	3,448	8.15	930	141	22,078	24.70	3.63	4.28
108	79.7	13.2	41.44	"	26,895	10.83	780	3,940	6.83	895	162	20,779	24.69	4.33	5.29
116	120.1	20.0	31.33	"	27,856	11.22	807	3,516	7.92	975	187	14,813	23.43	3.54	4.38
115	120.6	20.0	33.96	"	29,900	12.04	867	4,041	7.40	1036	187	15,883	23.74	3.84	4.69
102	160.3	26.6	22.16	"	24,410	9.83	707	3,271	7.46	803	188	8,663	24.78	4.00	5.22
105	157.6	26.2	28.03	"	27,513	11.08	797	3,829	7.19	951	260	9.929	23.73	3.96	5.43
113	158.7	26.4	30.12	"	28,427	11.45	824	4,001	7.11	968	206	10,835	24.17	4.07	5.17
106	160.9	26.7	32.91	"	30,943	12.47	897	4,627	6.69	1050	276	10,863	24.15	4.34	5.89
117	160.6	26.7	35.30	Partial.	30,747	12.39	891	4,163	7.39	1024	248	10,902	24.69	4.00	5.28
101	160.5	26.7	42.14	"	28,203	11.36	817	4,252	6.63	852	203	9,118	27.30	4.93	6.48
104	160.8	26.7	45.09	"	28,424	11.45	824	3,882	7.32	803	207	8,366	29.19	4.77	6.44
114	160.8	26.7	52.05	"	28,583	11.51	828	3,723	7.68	682	170	7,182	34.62	5.40	7.19

The Boiler Pressure was from 172 to 203 Pounds per Square Inch.

sure (205 pounds) was available as mean effective pressure at starting. On this basis the ratio of weight on drivers to maximum tractive effort was 4.35 : 1.

A summary of the results obtained from this locomotive is given in Table 1.

10. *Tests of Consolidation Locomotive, Lake Shore and Michigan Southern Railway Company.*—The second locomotive tested was No. 734, a two-cylinder simple locomotive, owned by the Lake Shore & Michigan Southern Railway Company, and built at the Brooks Locomotive Works. It was of the 2-8-0 type and known as class B-1, according to the railroad company's classification.

Twenty-one tests were made, the first on July 2 and the last on August 2. The total number of working days consumed in making these tests was twenty-nine, twelve of which were lost on account of difficulties experienced with the plant and six days on account of difficulties with the locomotive.

The principal dimensions and the details of the locomotive are shown in the following table:

Total weight, pounds	181,300
Weight on drivers, pounds	162,600
Cylinders (simple), inches	21 x 30
Diameter of drivers, inches	63
Fire-box heating surface, sq. ft.	218.92
Heating surface in tubes (water side), sq. ft.	2,638.97
Total heating surface (based on water side of tubes), sq. ft.	2,857 89
* Total heating surface (based on fire side of tubes), sq. ft.	2,541.22
Grate area, sq. ft.	33.76
Boiler pressure, pounds	200
Valves	Allen-Richardson
Link motion	Stephenson
Fire-box, Type	Narrow, on top of frames
Number of tubes	338
Outside diameter of tubes, inches	2
Length of tubes, inches	178.94

* Used in calculations.

The maximum tractive effort was 33,616 pounds, which was calculated on the assumption that 80 per cent. of the boiler pressure (200 pounds) was available as mean effective pressure at starting. On this basis the ratio of weight on drivers to maximum tractive effort was 4.84 : 1.

A summary of the results obtained from the locomotive is given in Table 2.

TABLE 2.

SUMMARY OF DATA. LAKE SHORE AND MICHIGAN SOUTHERN LOCOMOTIVE NO. 734.

Built by Brooks Locomotive Works, Dunkirk, N. Y., 1900.

	RUNNING CONDITIONS.				BOILER PERFORMANCE.					POWER.			WATER AND FUEL CONSUMPTION. POUNDS.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
201	40.3	7.6	19.1	Full.	11,122	4.38	322	1,109	10.0	299	67	11,531	29.56	3.53	4.55
202	40.5	7.6	30.7	"	15,162	5.97	439	1,412	10.7	434	33	19,860	27.78	3.10	3.36
203	40.1	7.5	41.3	"	18,599	7.32	539	2,027	9.2	550	59	24,522	27.31	3.55	3.99
204	80.0	15.0	43.9	"	28,500	11.21	826	4,468	6.3	901	84	20,444	26.11	4.88	5.39
205	80.5	15.1	17.3	"	16,864	6.64	489	1,781	9.5	527	64	11,515	25.89	3.28	3.73
206	80.1	15.0	30.7	"	23,580	9.28	683	2,729	8.6	783	51	18,288	24.55	3.41	3.65
208	80.0	15.0	40.7	"	29,453	11.59	854	4,152	7.1	962	68	22,371	24.93	4.22	4.54
209	159.3	29.9	21.1	"	26,780	10.54	776	3,435	7.7	866	80	9,861	25.11	3.86	4.25
210	159.9	30.0	23.3	"	28,713	11.30	832	4,096	7.0	954	103	10,642	24.67	4.22	4.72
211	160.0	30.0	29.0	"	31,361	12.34	909	5,061	6.2	995	134	10,755	26.04	5.02	5.80
212	160.3	30.0	27.4	"	30,620	12.05	887	4,695	6.5	1054	157	11,188	23.92	4.38	5.16
213	160.4	30.1	39.8	Partial.	29,710	11.69	861	4,505	6.6	887	85	9,998	27.64	5.01	5.54
214	39.6	7.4	19.4	Full.	......	......	...	......	.....	306	54	12,725	.....	....	....
215	80.1	15.0	19.7	"	......	......	...	......	.....	569	94	11,847	.....	....	....
216	79.5	14.9	40.1	"	......	......	...	......	.....	995	81	22,001	.....	....	....
217	159.2	29.8	38.5	Partial.	29,190	11.49	846	3,975	7.3	799	40	9,540	30.11	4.91	5.17
218	160.1	30.0	39.0	"	29,352	11.55	851	4,509	6.5	865	22	10,538	28.01	5.14	5.28
219	158.8	29.8	30.9	"	28,712	11.29	832	4,079	7.0	923	86	10,546	25.53	4.34	4.78
220	160.0	30.0	24.6	"	28,280	11.13	820	3,950	7.2	942	135	10,096	24.67	4.12	4.81
221	118.9	22.3	35.7	Full.	......	......	...	......	.....	1098	141	16,104	.....	....	....
222	119.4	22.4	37.5	"	......	......	...	......	.....	1098	98	16,744	.....	....	....

The Boiler Pressure was from 178 to 204 Pounds per Square Inch.

11. *Tests of Consolidation Locomotive, Michigan Central Railroad Company.*—The third locomotive tested was No. 585, owned by the Michigan Central Railroad Company and built by the American Locomotive Company at its Schenectady Works. It was of the 2-8-0 type and known as class W, according to the railroad company's classification. This locomotive was a two-cylinder cross compound.

It was on the plant from August 10 to August 27; work on the dynamometer consuming the time from August 3 to 10. In the twenty-two working days fourteen tests were made, twelve days being lost on account of the plant and three days on account of the locomotive.

The principal dimensions and details of the locomotive are shown in the following table:

Total weight, pounds	189,000
Weight on drivers, pounds	164,500
Cylinders (compound), inches	23 & 35 x 32
Diameter of drivers, inches	63
Fire-box heating surface, sq. ft.	165.69
Heating surface in tubes (water side), sq. ft.	3,015.34
Total heating surface (based on water side of tubes), sq. ft.	3,181.03
* Total heating surface (based on fire side of tubes), sq. ft.	2,819.20
Grate area, sq. ft	49.43
Boiler pressure, pounds	210
Valves, high pressure piston ; low pressure	Allen Richardson
Link motion	Stephenson
Fire-box, type	Radical stay, wide
Number of tubes	383
Outside diameter of tubes, inches	2
Length of tube, inches	190.38

* Used in calculations.

The maximum tractive effort was 45,613 pounds working simple and 31,823 pounds working compound. The ratio of weight on drivers to maximum tractive effort when working simple was 3.61 : 1 and when working compound, 5.17 : 1.

A summary of the results obtained from this locomotive is given in Table 3.

TABLE 3.

SUMMARY OF DATA. MICHIGAN CENTRAL LOCOMOTIVE NO. 585.

Built by the American Locomotive Co., Schenectady, N. Y., 1902.

Number of Test.	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke. H. P. C.	Throttle Opening.	Equivalent Evaporation. Pounds Per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
301	40.0	7.5	43.1	Full.	11,180	3.97	324	1,027	10.89	442	31	20,605	20.20	2.22	2.38
302	40.0	7.5	45.3	"	11,966	4.25	347	1,013	11.82	477	35	22,149	20.09	2.03	2.19
303	40.0	7.5	48.6	"	12,884	4.57	373	1,073	12.01	512	30	24,105	20.27	2.01	2.14
305	80.0	15.0	45.7	"	20,205	7.17	586	1,770	11.41	841	67	19,393	19.54	2.05	2.23
306	80.2	15.0	42.2	"	18,178	6.45	527	1,683	10.80	735	59	16,879	20.15	2.23	2.42
308	80.0	15.0	52.8	"	22,476	7.97	651	2,087	10.77	932	61	21,815	19.69	2.19	2.34
309	80.0	15.0	57.5	"	25,501	9.05	739	2,559	9.97	1041	61	24,539	20.03	2 41	2.56
311	117.9	22.1	50.6	"	......	......	...	.....	.....	998	97	15,297	.....	.....	....
312	160.0	30.0	49.6	"	25,075	8.89	727	2,687	9.24	890	160	9,138	23.18	2.96	3.61
313	160.0	30.0	50.7	"	27,576	9.78	793	2,767	9.97	992	168	10,308	22.80	2.74	3.30
316	160.0	30.0	64.1	"	29,700	10.53	861	3,876	7.66	1001	142	10,762	24.43	3.82	4.44
317	160.0	30.0	51.5	Partial.	24,538	8.70	711	2,444	10.04	910	183	9,102	21.98	2.62	3.28
318	160.0	30.0	60.3	"	26,864	9.53	779	2,738	9.81	938	176	9,530	23.56	2.87	3.54
319	160.0	30.0	66.0	"	27,370	9.71	793	2,638	10.38	934	196	9,236	24.14	2.79	3.52

The Boiler Pressure was from 175 to 211 Pounds per Square Inch.

12. *Tests of "Santa Fe" Type Locomotive, Atchison, Topeka & Santa Fe Railway System.*—The fourth locomotive tested was No. 929, owned by the Atchison, Topeka & Santa Fe Railway System and built at the Baldwin Locomotive Works. It was of the 2-10-2 type and known as class 900, according to the railroad company's classification. It was a four-cylinder compound.

This locomotive occupied the time from August 28 to September 17. In these twenty-one working days nine tests were run, the throttling tests being omitted as it was decided that the information would not be valuable on compound locomotives.

Four days were lost on account of difficulties with the plant, one on account of difficulties with the locomotive and eight and one-half days due to trouble experienced in getting the locomotive to the plant.

An accident reduced the number of available brakes to seven, which was unfortunate, as this locomotive was very powerful, and, with the low and varying water pressure in the mains during the period of the tests, it was possible to make tests only at powers below the full capacity of the locomotive. Its calculated tractive power was 63,612 pounds, and the highest draw-bar pull obtained in any test was 32,532 pounds.

Tests were only run at 40, 60 and 80 revolutions per minute. They were not, therefore, complete or conclusive, as the limits of the boiler capacity could not be ascertained.

The principal dimensions and the details of the locomotive are shown in the following table:

Total weight, pounds	285,740
Weight on drivers, pounds	233,760
Cylinders (compound), inches	19 & 23 x 32
Diameter of drivers, inches	56.5
Fire-box heating surface, sq. ft	216.36
Heating surface in tubes (water side), sq. ft	4,601.00
Total heating surface (based on water side of tubes), sq. ft	4,817.36
* Total heating surface (based on fire side of tubes), sq. ft	4,306.13
Grate area, sq. ft	58.41
Boiler pressure, pounds	225
Valves	Piston
Link motion	Stephenson
Fire-box, type	Radical Stay
Number of tubes	393
Outside diameter of tubes, inches	2.25
Length of tubes, inches	238.5

* Used in calculations.

The maximum tractive effort was 73,177 pounds working simple and 63,612 pounds working compound. The ratio of weight on drivers to maximum tractive effort, working simple, was 3.19 : 1 and when working compound, 3.67 : 1.

A summary of results obtained from this locomotive is given in Table 4.

TABLE 4.

SUMMARY OF DATA. ATCHISON, TOPEKA AND SANTA FE LOCOMOTIVE NO. 929.

Built by the Baldwin Locomotive Works, Philadelphia, Pa., 1903.

	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke. H. P. C.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
401	40.0	6.7	26.6	Full.	12,926	3.00	375	1,111	11.63	392	57	18,680	26.47	2.73	3.20
402	40.0	6.7	33.9	"	15,705	3.65	455	1,465	10.72	511	67	24,784	24.80	2.78	3.20
403	40.0	6.7	40.8	"	18,273	4.24	530	1,751	10.43	634	76	31,131	23.38	2.69	3.06
405	80.0	13.4	28.8	"	18,414	4.28	534	1,666	11.05	631	122	14,224	23.67	2.57	3.19
407	80.0	13.4	41.4	"	27,901	6.48	809	2,651	10.52	1089	124	26,929	20.98	2.39	2.70
408	81.3	13.6	51.4	"	36,813	8.55	1067	4,299	8.56	1258	120	31,240	24.04	3.37	3.74
410	60.0	10.1	26.1	"	16,350	3.80	474	1,437	11.38	511	101	15,285	25.80	2.73	3.39
411	60.0	10.1	33.7	"	20,195	4.69	585	1,935	10.44	705	107	22,279	23.22	2.68	3.16
412	60.6	10.2	41.9	"	23,814	5.53	690	2,381	10.00	889	101	29,005	21.84	2.63	2.96

The Boiler Pressure was from 213 to 217 Pounds per Square Inch.

13. *Tests of De Glehn Atlantic Type Locomotive, Pennsylvania Railroad Company.*—The fifth locomotive tested was No. 2,512, owned by the Pennsylvania Railroad Company and built from the designs of Messrs. De Glehn and Du Bosquet, by the Société Alsacienne de Constructions Mécaniques at Belfort, France. The locomotive was a four-cylinder balanced compound of the 4-4-2 type, and being the only locomotive of this type in this country, it has not been classified by the railroad company.

It was, with the exception of a few unimportant modifications, an exact duplicate of a number of locomotives furnished to the Northern Railway of France by the same builders. It was the only one tested having Serve ribbed tubes in the boiler.

In the "De Glehn" type the low-pressure cylinders are between the frames, and the high-pressure cylinders are outside the frames. The low-pressure cylinders are side by side and drive inside cranks set quartering on the forward driving axle. The high-pressure cylinders connect with outside crank pins in the drivers of the second driving axle.

The high-pressure cylinders are placed back on the frames in relation to the low-pressure cylinders, so that the main rods of the former are but $5\frac{1}{2}$ inches longer than those of the latter.

This locomotive had a separate valve gear for both high-pressure and low-pressure cylinders, and the cut-off in the low-pressure cylinders could be varied independently of the high-pressure cut-off.

Both high-pressure cylinders exhausted into a combined receiver and steam chest of 14.5 cubic feet capacity. A variable nozzle having an area of 17.5 to 43.5 square inches and operated from the cab controlled the exhaust blast and was varied to suit the points of cut-off which were used.

When operated simple, the high-pressure cylinders exhausted to the atmosphere and an auxiliary throttle admitted live steam to the low-pressure cylinders.

The front flue sheet was made of steel about one inch thick and the back sheet was copper about 1 5-16 inches thick.

The fire-box was made of copper, both sides and crown sheet being a single piece $\frac{5}{8}$ of an inch thick. The four top rows of staybolts were manganese bronze, the others copper, and all were drilled.

This locomotive was placed on the testing plant twice, the first period being from September 15 to October 8, the second period being from November 25 to December 3, a total of thirty days, during which time ten tests were made. In the first period of twenty-one days six tests were obtained; the most serious delay being due to parts of the locomotive running hot when high speeds were attempted.

Seven and one-half days were lost on account of heating of rod brasses, chiefly at the back end of the low-pressure main rod journals. The low-pressure rods were on the inside cranks where the brasses were necessarily narrow, and hence the pressure per unit of area was high.

The whole locomotive was unusually steady at all speeds, having very little motion of any kind.

On October 7, when the test at 320 revolutions was attempted,

the left front driving box ran hot after seven minutes, and it was necessary to stop. The construction of this locomotive was such that it was impossible to examine and repack the driving-box collars without dropping the driving wheels. Not having any drop pit, this was impossible at the testing plant.

The locomotive was sent to the Terre Haute shops of the Vandalia Line, repaired and run on the road until it was thought to be in good condition, and was then returned to St. Louis.

After tests on the Atchison, Hanover and New York Central locomotives were completed, the De Glehn was again placed on the testing plant on November 25 and was taken off on December 3. In this period of nine days, four tests were obtained. Considerable difficulty was encountered in getting the locomotive to steam, it having been impossible to run at cut-offs as long as those obtained on the road.

The coal used was soft and readily broke into small pieces. There was no shaking arrangement in the grates, and if the nozzle was decreased in size to increase the draft and clear the fire, there was a tendency to fill up the smoke-box with cinders. For these reasons the indicated horse-powers obtained were not as large as the maximum reported from road tests.

This locomotive will be very carefully tested on the testing plant as soon as it is erected at Altoona, with a view to developing the value of this system of compounding.

After November 29 no more tests were obtained, due to heating of the back ends of the main rods, and the locomotive was removed from the plant on December 3. In these twenty-six working days two and one-half days were lost on account of difficulties experienced with the plant and seventeen due to troubles with the locomotive.

The principal dimensions and the details of the locomotive are shown in the following table:

Total weight, pounds	164,000
Weight on drivers, pounds	87,850
Cylinders (compound), inches	$14\frac{3}{16}$ & $23\frac{11}{16}$ x $25\frac{1}{4}$
Diameter of drivers, inches	80
Fire-box heating surface, sq. ft.	177.28
Heating surface in tubes (water side), sq. ft.	1,468.87
Total heating surface (based on water side of tubes), sq. ft.	1,646.15
Total heating surface (based on fire side of tubes), sq. ft.	2,656.48
Grate area, sq. ft	33.39
Boiler pressure, pounds per sq. inch	225
Valves "D" Slide, H. P. balanced ; L. P. not balanced.	
Valve motion	Walschaert
Fire-box, type	Belpaire
Number of tubes (Serve)	139
Outside diameter of tubes, inches	$2\frac{3}{4}$
Length of tube, inches	176.14

The maximum tractive effort was 22,698 pounds working simple and 16,700 pounds working compound, which was calculated on the assumption that 80 per cent. of the boiler pressure (225 pounds) was available as mean effective pressure at starting. On this basis the ratio of weight on drivers to maximum tractive effort was 3.87 : 1 working simple and 5.26 : 1 working compound.

A summary of the results obtained from this locomotive is given in Table 5.

TABLE 5.

SUMMARY OF DATA. PENNSYLVANIA RAILROAD LOCOMOTIVE No. 2,512

Built by the Société Alsacienne de Constructions Mécaniques, Belfort, France, 1904.

	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke. H. P. C., L. P. C.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
501	80.0	19.1	26.9 52.1	Full.	8,416	3.17	244	690	12.19	310	33	5,443	21.20	2.09	2.34
502	80.0	19.1	39.1 60.0	"	11,580	4.36	336	1,005	11.53	496	56	8,615	18.60	1.94	2.19
505	160.0	38.3	25.2 52.3	"	13,089	4.93	379	1,157	11.31	524	81	4,343	19.95	2.12	2.52
506	160.3	38.3	27.3 52.7	"	13,810	5.20	400	1,259	10.97	524	71	4,448	21.15	2.31	2.67
507	160.0	38.3	38.4 60.1	"	20,091	7.56	582	2,247	8.94	809	199	5,976	19.60	2.69	3.57
508	160.0	38.3	49.7 69.8	"	23,999	9.04	696	3,038	7.90	945	102	8,262	20.67	3.14	3.52
510	240.0	57.4	27.7 50.0	"	16,273	6.13	472	1,751	9.24	597	243	2,309	21.95	2.86	4.83
511	240.0	57.4	29.8 57.2	"	18,933	7.13	549	2,395	7.90	653	92	3,664	22.69	3.52	4.10
512	240.0	57.4	34.2 62.2	"	22,088	8.32	640	2,641	8.36	802	149	4,268	21.62	3.17	3.90
513	280.0	67.0	29.2 57.9	"	23,453	8.83	680	2,897	8.10	682	172	2,857	27.05	4.10	5.48

The Boiler Pressure was from 206 to 220 Pounds per Square Inch.

14. *Tests of Atlantic Type Locomotive, Atchison, Topeka & Santa Fe Railway System.*—The sixth locomotive tested was No. 535, owned by the Atchison, Topeka & Santa Fe Railway System and was built at the Baldwin Locomotive Works. It was of the 4-4-2 type and known as class 507, according to the railroad company's classification.

This was a Vauclain four-cylinder compound locomotive of the type introduced by the Baldwin Locomotive Works. The two low-pressure cylinders were outside of the frames and the two high-pressure cylinders between them. All four cylinders were connected to the front axle. The high-pressure crossheads were connected to a cranked axle in which the crank pins were set quartering or 90 degrees apart. The low-pressure crossheads were connected to crank pins 90 degrees apart in the front drivers. On either side of the locomotive there was a high and low pressure cylinder connected to cranks set opposite or at 180 degrees to each other. The high and low pressure cylinders were in line across the locomotive, so that the high and low pressure connecting rods were of the same length.

The cut-off in the high and low-pressure cylinders could not be varied independently, as the valves for each set of high and low-pressure cylinders were actuated by a single valve gear.

The first official test on this locomotive was made on October 12. Locomotive No. 535 occupied the time from October 9 to November 1, a period of twenty-four days. During that period eleven tests were made, the work being delayed by hot inside crank brasses on the locomotive. Four days were lost by troubles due to the plant and eight on account of the locomotive.

On October 27 an attempt was made to run a test at 320 revolutions, but after two minutes the babbitt melted out of the inside crank brasses. New brasses were put in and this test was again tried the next day with the same result. As it appeared unlikely that a test could be run at 320 revolutions, a test at 280 revolutions was tried with success. On October 31 a test at 320 revolutions failed after 10 minutes for the same reason as the others, and on November 1 the same thing occurred again. It was then decided to remove the locomotive from the plant, as four attempts at tests had failed at this speed, and it was evidently impossible to run it cool at this speed on the testing plant.

This locomotive vibrated considerably at 240 revolutions, the movement of the pilot being about eight-tenths of an inch. The

wires that were run under the wheels, to determine the effect of the counterbalance weights on rail pressure, showed that at 320 revolutions the driving wheel lifted from the supporting wheel a height of at least six-one-hundredths of an inch. A tendency to run to the right side was also noticed, the driving wheels bearing so hard against the right supporting wheels that the flanges of the drivers were badly cut. The locomotive was jacked over to the left, and wooden wedges driven in between the frame and trailer to hold it over, but the vibration of the locomotive soon loosened the wedges and the cutting of the flanges would commence again. The locomotive was squared, the wheels trammed, and the supporting wheels were correctly set. The circumference of the driving wheels on the right side was two-thousandths of a foot larger than on the left side.

The principal dimensions and the details of the locomotive are shown in the following table:

Total weight, pounds	201,500
Weight on drivers, pounds	99,200
Cylinders (compound), inches	15 & 25 x 26
Diameter of drivers, inches	79
Fire-box heating surface, sq. ft	220.3
Heating surface in tubes (water side), sq. ft	3,016.71
Total heating surface (based on water side of tubes), sq. ft	3,237.01
* Total heating surface (based on fire side of tubes), sq. ft	2,902.05
Grate area, sq. ft	48.36
Boiler pressure, pounds	220
Valves	Piston
Link motion	Stephenson
Fire-box, type	Wagon top
Number of tubes	273
Outside diameter of tubes, inches	2.25
Length of tube, inches	225.14

* Used in calculations.

The maximum tractive effort was 26,182 pounds working simple and 19,245 pounds working compound, which was calculated on the assumption that 80 per cent. of the boiler pressure (220 pounds) was available as mean effective pressure at starting. On this basis the ratio of weight on drivers to maximum tractive effort was 3.79 : 1 working simple and 5.15 : 1 working compound.

A summary of the results obtained from this locomotive is given in Table 6.

TABLE 6.

SUMMARY OF DATA. ATCHISON, TOPEKA AND SANTA FE LOCOMOTIVE No. 535.

Built by the Baldwin Locomotive Works, Philadelphia, Pa., 1904.

	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
601	80.0	18.8	26.7	Full.	10,656	3.67	309	875	12.17	356	53	6,058	23.67	2.34	2.75
602	80.0	18.8	31.0	"	13,012	4.48	377	1,169	11.13	479	86	7,847	21.67	2.35	2.86
603	80.0	18.8	37.6	"	15,638	5.39	453	1,381	11.33	570	69	9,998	21.91	2.34	2.66
604	80.0	18.8	53.0	"	20,713	7.14	600	2,058	10.06	808	166	12,815	20.56	2.48	3.12
605	160.0	37.6	36.1	"	21,128	7.28	612	2,055	10.28	877	122	7,535	19.44	2.28	2.65
606	160.0	37.6	43.0	"	24,885	8.58	721	2,468	10.09	1000	127	8,708	20.17	2.42	2.77
607	160.0	37.6	50.5	"	30,900	10.65	896	3,258	9.49	1296	181	11,119	19.41	2.47	2.87
609	239.9	56.4	46.4	"	34,668	11.95	1005	4.452	7.79	1414	392	6,803	19.99	3.10	4.29
610	240.0	56.4	52.9	"	39,539	13.62	1146	5,831	6.78	1549	245	8,679	20.82	3.72	4.41
611	240.0	56.4	51.3	"	40,964	14.11	1187	5,701	7.19	1621	352	8,444	20.48	3.47	4.43
613	280.0	65.8	47.7	"	37,463	12.91	1086	5,104	7.34	1460	562	5,120	20.73	3.45	5.60

The Boiler Pressure was from 216 to 222 Pounds per Square Inch.

15. *Tests of Atlantic Type Locomotive, Hannoversche Maschinenbau-Actien-Gesellschaft.*—The seventh locomotive tested was No. 628, built by the Hannoversche Maschienenbau-Actien-Gesellschaft, vormals Georg Egestorff, Linden vor Hannover, Germany, and was presented for test by the builders. This locomotive was built for the Hannover directorate of the Royal Prussian Railway Administration (Koeniglich Preuss. Eisenbahn Verwaltung, Direktion Hannover) and was delivered to them at the close of the Exposition.

The locomotive was a four-cylinder balanced compound with superheater, and was known as the S8 class, according to the railroad company's classification.

The valve motion was the Heusinger von Waldegg, otherwise known as the Walschaert modified by von Borries.

The four cylinders were set across the locomotives on the center line of the leading truck. The two high-pressure cylinders were between the frames and the two low-pressure cylinders were outside the frames. The high and low pressure cylinders of each

pair were cast in one piece with the corresponding steam chests, and the two groups of cylinders bolted together. The four cylinders were all connected to the forward axle. The cranks of the high-pressure cylinders, on one side, were set at 180 degrees to each other.

This locomotive was equipped with a Pielock superheater, which consisted of a chamber in the shell of the boiler, using a portion of the boiler tubes as superheating surface. It was located far enough from the fire-box so that the tubes could not be overheated. The main part of the superheater consisted of a box into which the ends of the boiler tubes were lightly rolled. This box was divided into compartments by plates parallel to the tubes, so as to get a long contact of the steam with the heating surface. The steam, at boiler pressure, passed into the superheater and then through the several compartments on its way to the cylinders.

This locomotive was on the testing plant from November 2 until November 12, a period of eleven days. In this time ten tests were made; all of the lost time, except half a day, being due to the locomotive.

The chief difficulty was caused by the inefficiency of the draft appliances. The fire did not burn evenly on the grate, being very intense next the flue sheet under the brick arch, and very dull and without draft near the fire-door.

There was no diaphragm plate or petticoat pipe in the front end, and the only way to adjust the draft was by changing the size of the nozzle. The height could not be changed, as no other exhaust nozzle pipes were on hand; the nozzle was made smaller; but this, of course, only increased the draft and did not make the fire burn more evenly.

The coal used was unlike that used in Germany. The locomotive was adjusted for burning the German fuel. In addition to the difficulty introduced by the different fuel, the locomotive was, undoubtedly, over-cylindered for American practice. This is clearly shown by the following comparisons of ratios:

Locomotive Number.	Heating Surface to Cylinder Volume.	Grate Surface to Cylinder Volume.
628 (Hannover)	120.64	2.00
* 628 "	101.10	2.00
3000 (N. Y. C.)	139.86	2.32
535 (A. T. & S. F.)	145.83	2.43
2512 (P. R. R.)	154.14	1.93
† E2a (P. R. R.)	236.51	5.69

* Superheating surface not included.

† This locomotive was not tested, but ratios are given as typical of a simple Atlantic type locomotive.

The tailpiece of the locomotive was made of a steel plate three-quarters of an inch thick, and covered the back of the frames, which were of the plate type. It was necessary to attach the fastenings for the dash-pots to this plate, and above 280 revolutions per minute it was not sufficiently stiff to prevent excessive vibration; the action of the dash-pots, therefore, did not protect the dynamometer.

The injectors on this locomotive did not admit of adjusting the feed as closely as American injectors do, so that the injectors had to be put on and off a great many times in the light tests. Consequently, the boiler pressure varied a good deal and considerable water was wasted at the injector overflow, but as the waste was measured it introduced no error.

The principal dimensions and the details of this locomotive are shown in the following table:

Total weight, pounds	133,350
Weight on drivers, pounds	65,350
Cylinders (compound), inches	$14\frac{5}{32}$ & 22 x $23\frac{5}{8}$
Diameter of drivers, inches	78
Fire-box heating surface, sq. ft	105.59
Heating surface in tubes (water side), sq. ft. not inc. superheater.	1,932.16
Total heating surface (based on water side of tubes), including superheater, sq. ft	1,753.15
* Total heating surface (based on fire side of tubes), including superheater, sq. ft	1,753.15
Grate area, sq. ft	29.06
Boiler pressure, pounds per square inch	200
Valves, high pressure piston; low pressure	Allan balanced
Valve motion	Von Borries simplified Heusinger von Waldegg
Fire-box, type	Wide
Number of tubes in boiler	241
Number of tubes in superheater	241
Outside diameter of tubes, inches	2
Length of tubes (not including superheater), inches	143.78
Length of tubes in superheater, inches	29.92

*Used in calculations.

The maximum tractive effort was 19,459 pounds working simple and 13,789 pounds working compound. The ratio of weight on drivers to maximum tractive effort when working simple was 3.36 : 1 and 4.74 : 1 working compound.

A summary of results of this locomotive is presented in Table 7.

TABLE 7.

SUMMARY OF DATA. HANNOVER LOCOMOTIVE No. 628.

Built by the Hannoversche Maschinenbau-Actien-Gesellschaft, Linden vor Hannover, Germany, 1904.

	Running Conditions.				Boiler Performance.					Power.			Water and Fuel Consumption. Pounds.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke. H. P. C.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour.	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Dry Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
701	80.0	18.6	35.2	Full.	9,560	5.45	277	997	9.57	376	22	7,136	18.09	2.44	2.60
702	80.0	18.6	44.9	"	12,000	6.85	348	1,206	9.95	480	34	9,016	17.82	2.34	2.52
705	160.0	37.1	37.6	"	14,720	8.40	427	1,521	9.68	623	73	5,552	16.81	2.27	2.57
706	160.1	37.1	43.2	"	16,696	9.52	484	1,832	9.11	729	83	6,516	16.60	2.38	2.69
707	160.0	37.1	47.8	"	19,789	11.29	574	2,679	7.39	814	59	7,622	17.86	3.15	3.40
708	160.0	37.1	47.4	"	19,725	11.25	572	2,653	7.49	801	139	6,690	18.16	3.21	3.88
709	240.0	55.7	35.3	"	14,620	8.34	424	1,592	9.18	631	93	3,624	16.67	2.38	2.80
710	239.4	55.6	38.8	"	16,670	9.51	483	2,165	7.70	710	87	4,203	17.36	2.96	3.37
711	240.0	55.7	46.4	"	20,834	11.88	604	3,523	5.91	816	172	4,339	18.80	4.19	5.31
712	280.3	65.0	35.8	"	19,913	11.36	577	2,525	7.89	688	95	3,422	21.29	3.53	4.09

The Boiler Pressure was from 187 to 204 Pounds per Square Inch.

16. *Tests of Atlantic Type Locomotive, New York Central and Hudson River Railroad.*—The eighth locomotive tested was No. 3,000, owned by the New York Central & Hudson River Railroad Company, and built by the American Locomotive Company at its Schenectady Works. It was of the 4-4-2 type and known as the I1 class, according to the railroad company's classification. It was a four-cylinder balanced compound, designed by Mr. F. J. Cole.

This locomotive had two high-pressure cylinders between the frames, set a little forward of the smoke-box. These high-pressure cylinders were connected to the forward axle, the cranks of which were set quartering. The low-pressure cylinders were outside the frames and connected to the second driving axle. Each outside crank pin was set 180 degrees with its adjacent inside crank. This arrangement made necessary the use of shorter connecting rods on the inside than on the outside.

The cut-off in the high and low-pressure cylinders could not be varied independently, as the valve for each set of high and low-pressure cylinders was actuated by a single valve gear.

The steam for the high-pressure cylinders passed through an opening in the saddle to a short pipe entering the top of the steam chest.

When working simple a starting valve operated from the cab admitted live steam, at a reduced pressure, to the low-pressure cylinders.

The "Perfection" fuel economizer, consisting of special fire-doors and means for admitting air above the fire, was used on this locomotive.

The locomotive was placed on the plant November 13 and removed November 24. During these twelve days eleven tests were run. The last test was at a speed of 320 revolutions or 75 miles an hour and lasted for one hour. It was not possible to run any of the other locomotives at that speed for that length of time.

It ran very steadily and showed excellent counterbalancing in every way.

The principal dimensions and the details of the locomotive are shown in the following table:

Total weight, pounds	200,000
Weight on drivers, pounds	110,000
Cylinders (compound), inches	15½ & 26 x 26
Diameter of drivers, inches	151.69
Heating surface in tubes (water side), sq. ft.	3,255.27
Total heating surface (based on water side of tubes), sq. ft.	3,406.96
* Total heating surface (based on fire side of tubes), sq. ft.	3,000.05
Grate area, sq. ft.	49.90
Boiler pressure, pounds	220
Valves	Piston
Link motion	Stephenson
Fire-box, type	Wide
Number of tubes	390
Outside diameter of tubes, inches	2
Length of tubes, inches	191.29

* Used in calculations.

The maximum tractive effort was 27,890 pounds working simple and 20,590 pounds working compound, which was calculated on the assumption that 80 per cent. of the boiler pressure (220 pounds) was available as mean effective pressure at starting. On

this basis the ratio of weight on drivers to maximum tractive effort was 3.94 : 1 working simple and 5.34 : 1 working compound.

A summary of the results of this locomotive is presented in Table 8.

TABLE 8.

SUMMARY OF DATA. NEW YORK CENTRAL LOCOMOTIVE NO. 3,000.

Built by the American Locomotive Co., Schenectady, N. Y., 1904.

	RUNNING CONDITIONS.				BOILER PERFORMANCE.					POWER.			WATER AND FUEL CONSUMPTION. POUNDS.		
Number of Test.	Revolutions per Minute.	Miles per Hour.	Cut off. Per cent. of Stroke. H. P. C.	Throttle Opening.	Equivalent Evaporation. Pounds per Hour	Equivalent Evaporation per Square Foot of Heating Surface per Hour. Pounds.	Boiler Horse-power.	Pounds of Dry Coal per Hour.	Equivalent Evaporation per Pound of Coal.	Indicated Horse-power.	Machine Friction Horse-power.	Dynamometer Pull. Pounds.	Steam per I. H. P. per Hour. Pounds.	Coal per I. H. P. per Hour. Pounds.	Coal per D. H. P. per Hour. Pounds.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
801	79.8	18.7	36.0	Full.	15,036	5.01	436	1,287	11.7	567	179	7,781	20.78	2.17	3.17
802	80.0	18.8	45.9	"	18,703	6.23	542	1,749	10.7	714	108	12,121	20.47	2.34	2.75
805	160.0	37.5	36.3	"	24,040	8.01	697	2,357	10.2	967	72	8,940	19.60	2.34	2.53
806	160.0	37.5	43.7	"	31,142	10.38	903	3,113	10.0	1253	76	11,766	19.95	2.44	2.60
807	160.0	37.5	57.1	"	39,872	13.29	1156	4,880	8.2	1490	212	12,780	21.57	3.23	3.77
809	240.0	56.3	32.2	"	29,960	9.99	868	3,024	9.9	1143	179	6,422	21.05	2.60	3.08
811	240.0	56.3	46.6	"	44,765	14.92	1297	5,802	7.7	1630	160	9,796	22.18	3.52	3.90
812	240.0	56.3	53.7	"	49,025	16.34	1421	6,694	7.3	1641	166	9,831	24.14	4.04	4.49
813	280.1	65.7	32.2	"	33,055	11.02	958	3,475	9.5	1192	223	5,530	22.27	2.87	3.53
814	280.0	65.7	38.2	"	37,721	12.57	1093	3,889	9.7	1369	180	6,788	22.19	2.80	3.23
815	320.0	75.0	41.0	"	38,973	12.99	1130	4,928	7.9	1336	290	5,224	23.51	3.64	4.65

The Boiler Pressure was from 209 to 222 Pounds per Square Inch.

A SUMMARY OF CONCLUSIONS.

17. *Boiler Performance.*—1. Contrary to a common assumption, the results show that when forced to maximum power the large boilers delivered as much steam per unit area of heating surface as the small ones.

2. At maximum power, a majority of the boilers tested delivered 12 or more pounds of steam per square foot of heating surface per hour; two delivered more than 14 pounds; and one, the second in point of size, delivered 16.3 pounds. These values

expressed in terms of boiler horse-power per square foot of heating surface are 0.34, 0.40 and 0.47, respectively.

3. The two boilers holding the first and second place with respect to weight of steam delivered per square foot of heating surface are those of passenger locomotives.

4. The quality of steam delivered by the boilers of locomotives under constant conditions of operation is high, varying somewhat with different locomotives and with changes in the amount of power developed between the limits of 98.3 per cent. and 99.0 per cent.

5. The evaporative efficiency is generally maximum when the power delivered is least. Under conditions of maximum efficiency most of the boilers tested evaporated between 10 and 12 pounds of water per pound of dry coal. The efficiency falls as the rate of evaporation increases. When the power developed is greatest its value commonly lies between limits of 6 and 8 pounds of water per pound of dry coal.

6. The observed temperature of the fire-box under low rates of combustion lies between the limits of 1,400 degrees Fahr. and 2,000 degrees Fahr., depending apparently upon characteristics of the locomotive. As the rate of combustion increases the temperature slowly increases, maximum values generally lying between the limits of 2,100 and 2,300 degrees Fahr.

7. The smoke-box temperature for all boilers, when working at light power, is not far from 500 degrees Fahr. As the power is increased the temperature rises, the maximum value depending upon the extent to which the boiler is forced. For the locomotives tested it lies in most cases between 600 and 700 degrees.

8. With reference to grate area, the results prove beyond question that the furnace losses due to excess air are not increased by increasing the area. In general, it appears that the boilers for which the ratio of grate surface to heating surface is largest are those of greatest capacity.

9. A brick arch in the fire-box results in some increase in furnace temperature and improves the combustion of the gases.

10. The loss of heat through imperfect combustion is in most cases small, except as represented by the discharge from the stack of solid particles of fuel.

11. Relatively large fire-box heating surface appears to give no advantage either with reference to capacity or efficiency. The fact

seems to be that the tube-heating surface is capable of absorbing such heat as may not be taken up by the fire-box.

12. The value of the Serve tube over the plain tube of the same outside diameter, either as a means for increasing capacity or efficiency, was not definitely determined.

13. The draft in the front end for any given rate of combustion, as measured in inches of water, depends upon the proportions of the locomotive and the thickness and condition of the fire. Under light power its value may not exceed an inch, but it increases rapidly as the power is increased. Representative maximum values derived from the tests lie between the limits of 5 inches and 8.8 inches.

14. Insufficient openings in the ash-pans and the mechanism of the front end, especially the diaphragm, are shown by the tests to lead to the dissipation of considerable portions of the draft force.

18. *The Engine.*—15. The indicated horse-power of the modern simple freight locomotive tested may be as great as 1,000 or 1,100; that of a modern compound passenger locomotive may exceed 1,600 horse-power.

16. The maximum indicated horse-power per square foot of grate surface lies, for the freight locomotives, between the limits of 31.2 and 21.1; for passenger locomotives, between the limits of 33.5 and 28.1.

17. The steam consumption per indicated horse-power hour necessarily depends upon the conditions of speed and cut-off. For the simple freight locomotives tested the average minimum is 23.7. The consumption when developing maximum power is 23.8, and when under those conditions, which proved to be the least efficient, 29.0.

18. The compound locomotives tested, using saturated steam, consumed from 18.6 to 27 pounds of steam per indicated horse-power hour. Aided by a superheater, the minimum consumption is reduced to 16.6 pounds of superheated steam per hour.

19. In general the steam consumption of simple locomotives decreases with increase of speed, while that of the compound locomotives increases. From this statement it appears that the relative advantages to be derived from the use of the compound diminish as the speed is increased.

20. Tests under a partially opened throttle show that when the degree of throttling is slight the effect is not appreciable. When the degree of throttling is more pronounced, the performance

is less satisfactory than when carrying the same load with a full throttle and a shorter cut-off.

19. *The Locomotive as a Whole.*—21. The percentage of the cylinder power which appears as a stress in the draw-bar diminishes with increase of speed. At 40 revolutions per minute the maximum is 94 and the minimum 77; at 280 revolutions per minute the maximum is 87 and the minimum 62.

22. The loss of power between the draw-bar and the cylinder is greatly affected by the character of the lubricant. It appears from the tests that the substitution of grease for oil upon axles and crank pins increases the machine friction from 75 to 100 per cent.

23. The coal consumption per dynamometer horse-power hour for the simple freight locomotives tested is, at low speeds, not less than 3.5 pounds nor more than 4.5 pounds, the value varying with the running conditions. At the highest speeds covered by the tests the coal consumption for the simple locomotives increased to more than 5 pounds.

24. The coal consumption per dynamometer horse-power hour for the compound freight locomotives tested is, for low speeds, between 2.0 and 3.7 pounds. Results at higher speeds were obtained only from a two-cylinder compound, the efficiency of which under all conditions is shown to be very high. The coal consumption per dynamometer horse-power hour for this locomotive at the higher speed increases from 3.2 to 3.6 pounds.

25. The coal consumption per dynamometer horse-power hour for the four compound passenger locomotives tested varies from 2.2 to more than 5 pounds per hour, depending upon the running conditions. In the case of all of these locomotives the consumption increases rapidly as the speed is increased.

26. A comparison of the performance of the compound freight locomotives with that of the simple freight locomotives is very favorable to the compounds. For a given amount of power at the draw-bar the poorest compound shows a saving in coal over the best simple which will average above 10 per cent., while the best compound shows a saving over the poorest simple which is not far from 40 per cent. It should be remembered, however, that the conditions of the tests, which provide for the continuous operation of the locomotives at constant speed and load throughout the period covered by the observations, are all favorable to the compound.

27. It is a fact of more than ordinary significance that a steam

locomotive is capable of delivering a horse-power at the draw-bar upon the consumption of but a trifle more than 2 pounds of coal per hour. This fact gives the locomotives high rank as a steam-power plant.

28. It is worthy of mention that coal consumption per horse-power hour developed at the draw-bar by the different locomotives tested presents marked differences. Some of these are easily explained from a consideration of the characteristics of the locomotives involved. Where the data is not sufficient to permit the assignment of a definite cause there can be no doubt but that an extension of the study already made will serve to reveal it.

W. F. M. G.
E. M. H.
J. E. S.

DISCUSSION.

E. G. Bailey.—The locomotive tests as conducted by the Pennsylvania Railroad System are very interesting and thorough, yet there are one or two points that might be brought out in connection with the fuel and boiler part that seem to be neglected generally by mechanical men, namely, the sampling and analysis of coal. This may seem to be purely a chemist's work, and as far as his part of it is concerned it is usually well done, but the error comes in interpreting the results. In this report there are given ninety-seven proximate analyses and calorimeter determinations of separate samples of coal from the same mine. This particular coal was selected because of its low ash and good quality. It comes from a seam that produces a uniform quality, and being friable, it is much easier to obtain good samples than from the harder coals, as are most of them found on the western market. With these conditions in favor of uniform sampling the analyses show a variation of 7 per cent. in Ash and 12 per cent. in B.t.u. Errors such as these affect the boiler efficiency as calculated; also the various items of the heat balance. But, had the analyses been averaged as a whole or in groups, much more uniform and reliable results would have been obtained, as here given:

AVERAGE OF SAMPLES TAKEN ON EACH SERIES OF TESTS BY PENNSYLVANIA R.R. SYSTEM.

TABLE 9.

Test Series.	Number Samples.	Moisture.	Volume.	F. C.	Ash.	Sulphur.	B.t.u.
100......	17	1.02	16.24	74.91	7.83	.89	14,037
200......	16	.95	16.65	76.10	6.30	.91	14,736
300......	13	.98	17.14	75.78	6.10	1.05	14,765
400......	9	.87	17.37	75.53	6.23	1.19	14,876
500......	10	.96	16.97	75.76	6.31	1.01	14,773
600......	11	1.09	16.96	75.68	6.27	1.18	14,802
700......	10	1.09	16.46	76.20	6.25	.79	14,835
800......	11	.97	16.46	76.27	6.30	.86	14,844
All......	97	.99	16.75	75.33	6.53	.98	14,621
except100	80	.99	16.86	75.90	6.25	1.00	14,804

Neglecting 1st set (100) Maximum difference from average.................................=0.11% +0.49%

Any one set of these analyses, except the first, which seems to be considerably different, could have been used in calculation with an error of not more than 0.1 per cent. in ash and 0.5 per cent. in B.t.u. from the true average of all samples. The heat units varying so much more than the ash is no doubt due to the calorimeter used in making these determinations, as the Carpenter is generally considered less reliable than the Mahler bomb. The following data illustrates this difference and shows how sets of analyses agree on the average:

TABLE 10.

No. Samples.	Moisture.	Volume.	F. C.	Ash.	Sulphur.	B.t.u.
30.....	1.01	19.24	71.32	8.43	1.29	14,370
43.....	.98	19.23	71.40	8.39	1.27	14,370
40.....	1.00	18.86	71.65	8.49	1.27	14,353
38.....	.99	19.14	71.38	8.49	1.25	14,376
43.....	.99	19.48	70.79	8.74	1.23	14,362
194.....	.99	19.19	71.31	8.51	1.26	14,366

Maximum variation from average............ +0.23% =0.09%

These samples were taken from barges of coal similar to that used on the Pennsylvania tests, but it comes from a mixture of four mines, and each average here given represents about 35,000

tons. The B.t.u. were determined by the Mahler bomb and agree within one-quarter of one per cent., which is sufficiently close for any work. Selecting samples from the Pennsylvania tests that show the greatest variation in ash:

TABLE 11.

Test.	Moisture.	Volume.	F. C.	Ash.	Sulphur.	B.t.u.
116....	.90	16.20	71.55	11.35	1.06	13,769
701....	1.00	16.49	78.20	4.31	.92	15,082
Difference....				7.04%		

While the greatest difference in heating value is found in other samples:

TABLE 12.

Test.	Moisture.	Volume.	F. C.	Ash.	Sulphur.	B.t.u.
220....	.73	16.75	76.98	5.54	.92	15,076
412....	.84	17.40	76.32	5.24	1.08	15,076
102....	.84	16.02	71.86	11.28	.77	13,315
109....	.84	16.02	71.86	11.28	.77	13,315
Difference....				6.04%	....	1,761

or 12.05 per cent. on a basis of 14,621 as the average of all, and a difference of 6.42 per cent. in the British Thermal Units per pound of combustible is found between tests 101 and 401.

It is highly improbable that the coal varied so much as this, but the trouble is in getting a correct sample, and in no case can one sample, taken in the ordinary way, be relied upon to represent even one ton of coal. The Testing Department of the Fairmount Coal Company have a great many series of results that demonstrate this, one of which is given below. These sixteen samples were taken from about three tons out of a car of coal as it was being wheeled in for burning on a boiler test. After loading the wheelbarrow, sixteen shovelfuls were taken from the pile and one put into each of the sixteen barrels. This made about 125-pound sample in each barrel, which was broken up so that it would pass through a $\frac{3}{4}$-inch screen, and quartered down in the usual way to about five pounds, which was crushed and ground up in the Laboratory, and each analyzed separately.

TABLE 13.

Lab. No.	Moisture.	Volume.	F. C.	Ash.	Sulphur.
7270.....	.90	16.56	72.86	9.68	.79
7271.....	1.06	16.44	72.22	10.28	.77
7272.....	1.02	16.96	68.10	13.92	.94
7273.....	1.00	16.42	71.36	11.22	.89
7274.....	.92	16.00	72.20	10.88	.84
7275.....	.96	17.08	72.16	9.80	.85
7276.....	.86	16.52	70.78	11.84	.79
7277.....	.92	15.80	73.00	10.28	.89
7278.....	1.04	15.18	73.68	10.10	.84
7279.....	1.08	16.36	71.92	10.64	.70
7280.....	1.12	16.24	72.58	10.06	.90
7281.....	1.00	16.18	72.10	10.72	.93
7282.....	1.00	17.30	72.24	9.46	.85
7283.....	.92	16.46	72.96	9.66	.85
7284.....	1.00	15.56	72.36	11.08	1.05
7285.....	1.00	16.56	71.10	11.34	.85
Average..	.99	16.35	71.98	10.68	.86
Maximum Variation				4.46	
" " from average......				(+3.24 (−1.22	

From these and more elaborate results where as high as 100 samples have been taken in a similar way, we learn that any one sample may be as much as 5 per cent. above or 2 per cent. below the average in ash, depending upon the percentage and nature of impurities in the coal. The sulphur varies in about the same proportion. Any ten samples will give an average within 0.25 per cent. of that obtained from 20 or more samples. In no case where the analysis of any coal is desired within one per cent. should less than five samples be taken and analyzed separately. After a sample is pulverized and reduced to about 50 grams the variations in analysis are within 0.1 per cent.

If these facts were better recognized among mechanical men it would be a big step toward obtaining more consistent results in the boiler tests and correcting the erroneous data which is now so largely copied in most handbooks.

When Bulletin No. 4 was published I thought it would be interesting to work out heat balances from the data given, but instead of using each analysis of coal, they were all averaged and assumed to represent the coal used on each test, which gave more uniform results than if the separate analyses had been taken:

TABLE 14.

ENGINE No. 1499.

PER CENT. OF HEAT REPRESENTED BY.

No. Test.	Latent Heat of Moisture Formed.	Products of Comb.	Air Excess.	C.O.	Cinders.	Sparks.	Evaporation.	Radiation and Undetermined.
110....	2.6	9.3	4.5	.3	5.0	1.9	78.9	2.5
111....	2.6	10.1	3.5	.3	2.5	1.0	76.5	3.5
103....	2.6	10.2	3.4	6.0	1.4	1.1	66.6	8.7
109....	2.6	10.2	3.5	4.6	3.7	.9	59.8	14.7
112....	2.6	11.7	1.7	9.4	5.6	.9	56.1	12.0
118....	2.6	12.1	2.3	14.3	5.6	1.1	55.5	6.5
108....	2.6	12.6	1.3	19.0	8.6	3.2	46.5	6.2
116....	2.6	12.0	2.2	11.1	8.8	3.5	54.0	6.8
115....	2.6	12.1	1.3	13.8	9.0	1.9	50.3	9.0
102....	2.6	11.5	2.6	4.4	6.4	1.5	50.9	20.1
105....	2.6	12.0	3.0	6.5	7.2	2.4	49.0	17.3
113....	2.6	12.8	2.5	10.5	.9	1.4	48.2	21.1
106....	2.6	13.7	4.0	7.0	...	...	45.5	27.2
117....	2.6	12.5	5.3	10.1	7.0	1.6	50.4	10.5
101....	2.6	12.1	2.5	10.4	7.6	3.2	45.8	15.8
104....	2.6	14.4	2.0	11.9	1.2	2.0	49.1	16.8
114....	2.6	13.1	2.3	12.2	5.0	.9	52.1	11.8

As all items for calculating a heat balance were given it was thought that the item "Radiation and Undetermined" would include the radiation and loss due to unburned coal from the ash pan. But in the complete report I learn that this item includes also the heat required to evaporate the water used to wet the coal, the sparks that could not be collected in the stack receptacle, and perhaps some error in flue gas analysis, as only three samples were taken on each test. As it is very difficult, or practically impossible, to collect all sparks coming from the stack with the high velocity, about the only way to determine their amount is, working on an ash basis, to know the total weight of ash fired into the firebox by knowing the weight of coal and the percentage of ash contained therein; then weighing the ashes from the pan and determining the per cent. of ash, or incombustible, which being deducted from the weight of ash in the coal gives the weight of ash passing through the flues. A good sample of this can easily be obtained, and from its analyses the total weight of sparks and weight of combustible can be determined. With careful sampling, this gave very consistent results on road tests, which were presented to the Society by Prof. E. A. Hitchcock.

The analyses of the products of combustion show that the ratio

of carbon to available hydrogen burned was considerably less than that given in the ultimate analysis, which shows that a large percentage of carbon was escaping unburned, and considerably more than what was collected from the stack. Due to this carbon escaping unburned formula 110 on page 137 is in error for calculating the loss due to the formation of CO, for "C" in the formula should be the per cent. of carbon to coal that was actually burned either as CO or CO_2.

Plots 110, 210, etc., are misleading, as they all show the boiler efficiency to drop off rapidly as the temperature of the firebox increases. The real cause for the lower efficiency is the higher rate of combustion, which also produces higher firebox temperature. And if tests were selected with varying furnace temperature and constant rate of combustion, the curve would probably rise as the temperature is increased.

No. 1110.*

SOME STEPPING STONES IN THE DEVELOPMENT OF A MODERN WATER-WHEEL GOVERNOR.

BY MARK A. REPLOGLE, AKRON, OHIO.
(Member of the Society.)

Introduction.—In the preparation of a paper to be presented to the Franklin Institute in the year 1897, I was permitted to discuss either "Speed Government in Water-power Plants," or "Water-wheel Governors." The former was chosen. The paper was read before the Institute on December 14, 1897, and was published in full in the Journal of the Franklin Institute, Volume CXLV., No. 2, February, 1898. It represented the gleanings from my eight years' experience while trying to successfully govern water powers, or rather turbine wheels.

Up to this time a number of different types of governors had been devised and put into service, but I had not made sufficient progress to warrant any conclusive statements concerning my developments. Another eight years has passed during which time much hard work has been done, and the results of this period are of a more definite character.

A mechanical governor has been developed. Some of its principles are unique, and some of its actions are remarkable. It is my intention to show some of the steps that led up to the development of this mechanism, as well as to describe the machine, and some of the principles that enter into its construction.

In the Franklin Institute paper I made the following statement: "A properly constructed governor must open the water-wheel gates as fast as gravity can follow with water; no faster. It must close the gates slow enough to insure safety to the penstock; no faster. It must be capable of stopping the gates at any degree of opening. It must be endowed with the relay principle adjusted to co-operate properly with the power storage.

* Presented at the Chattanooga Meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions.*

"It must not be a separate and independent feature of the plant, but must be made a part of the plant in an intelligent manner; and at best it is only one of the factors in the government of a water-power plant. It must be remembered that all the governor can do is to open or close the gates as the variations in speed require, and no water wheel can be governed successfully by varying the gate openings, unless the same principles are adhered to that make government in steam engines a success."

This very general statement pointed out the work to be accomplished. This paper, while adding some new requirements, will be devoted *mainly to showing how to obtain the desired results.*

1. A modern water-wheel governor must have four distinct elements: 1st, the speed-control element; 2d, the gate-moving element; 3d, the temporary speed-control influence; and 4th, the permanent speed-control influence.

2. The control element in a modern water-wheel governor is a centrifugal speed governor. Its function is to trip or put into action the gate or valve-operating apparatus. This element is affected by, 1st, variation in speed of the turbine; 2d, the temporary-control influence; and 3d, the permanent-control influence.

3. The gate-moving element provides the power necessary to operate the wheel gates, and can open or close them as necessity requires. This element may be a pawl and ratchet device, a clutch, a shifting belt, a hydraulic ram, or a variable speed transmission. This element can use any convenient power supply, but the control element must always be driven by the water wheel to be governed, if speed regulation is required.

4. The temporary-control influence is used to cause the governor to anticipate the effects of water added, so as to prevent the condition known as hunting or racing. This element may be of hydraulic or mechanical construction, or a combination of both. The permanent-control influence is for the purpose of running several power units in multiple, causing an equal distribution of load.

5. Before proceeding further, I wish to make the following statements: A good water-wheel governor must be able to move the gates slowly or rapidly, as necessity requires. This discrimination savors of good judgment. A good governor must always stop operating before the speed has reached normal. In fact, it ceases to operate at a speed that is farther from normal than that necessary to cause it to begin operating at the start. This is "judgment." A good governor always ceases to operate before

the speed has returned to normal; sometimes it ceases a shorter time before the speed is right; and again it ceases a longer time before the speed has exactly reached normal, but always just long enough ahead of time to allow the speed condition to be right after the inertia effects of the water, the new load conditions, and the new power conditions are balanced. This would be called an act of reason if performed by a man, a horse, or a dog. Is it less so if performed by a machine? The perfect governor can place the water-wheel gates in position ahead of time with a precision that

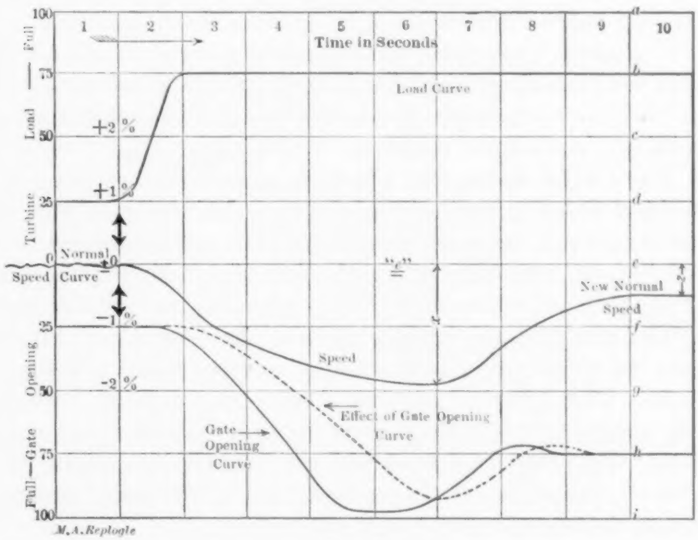

FIG. 1.

could not be reached by the designer of the machine if he had to do the governing by hand.

6. In order to pave the way to a fuller conception of the new governor, I can think of no better course than to show by some simple illustrations a few of the important stages in its evolution.

7. Before going into these details, it will be well to get an idea of the necessary action of a governor as compared with the speed and other conditions of the turbine and power-plant effects. I will, therefore, aim to show graphically in the diagram, Fig. 1, the relative time of the governor's actions as compared with the time of changes in load and speeds. A careful study of this diagram will reveal the fact that a good governor must be possessed with

powers that closely resemble reasoning. I will add further that the actual conditions as shown by the diagram were not understood in the earlier years of my governor work.

8. In order to make the necessary comparisons in Fig. 1, the ordinates 1, 2, 3, etc., divide time in seconds; the abscissas a, b, c, d, etc., represent load above the line e, and the gate opening below e. For further convenience e represents normal speed, while d and c represent 1 per cent. and 2 per cent. respectively above speed, and f and g represent 1 per cent. and 2 per cent. below normal speed. The load curve can be noted in the upper half and the gate-opening curve in the lower half of the diagram. X represents the temporary drop, and z represents the permanent drop in the governor's operation. Beginning at the left, it will be noted that up to the beginning of the second second the load is 25 per cent. of full capacity; the speed is zero or normal, and the gate opening is also 25 per cent. of full. At this point 50 per cent. of the full load is added, requiring a full second of time as shown. The speed begins to drop at the instant new load is added, but as it must vary enough to throw the governor into action, there is no gate movement until about the middle of the second. Then, as gravity must have time to overcome the inertia of more water, there is no power added until the beginning of the third second. (The dotted line shows the gate effect, which is always after the time of gate movement.) By the middle of the fifth second the gate is nearly open, and remains almost stationary for almost one second. The speed, however, keeps on falling until the beginning of the seventh second. (Please note that it is the *effect* of the gate opening that does the governing, and not the amount of the opening; also that the governor ceased to open the gate one and one-half seconds before the speed ceased dropping, and fully five seconds before the speed had risen to the new normal condition, which is one-half of 1 per cent. below the orginal normal speed.)

9. If the governor had kept opening the gate during all the time that the speed was below normal, the speed would rise far above normal by the time the full effect could be realized, and the result would be an over-running or hunting.

10. The momentum of the power unit has a great deal to do with the rapidity of drop in speed when a load is added, and the hydraulic conditions have a great deal to do with the difference in time between the gate movements and their effects. Both of these conditions, while they affect governing materially, are in no

way determined by the governor man. A governor must be capable of adjustment to the various turbine conditions, or failure will result.

11. Having shown what a good governor must be able to do, I will now proceed to show some of the steps leading up to such a machine, referring, of course, to my personal experience. My first governor patents contemplated a mercurial speed governor, but on account of being able to procure a very good mechanical speed governor to use as an indicator of speed conditions, the mercurial governor was never used. The majority of governors in use in the United States, in 1890, were of a simple two-stage type. That is, the movement of parts in the centrifugal element would trip or set into action the power, or gate-moving, elements. This class of governors would continue to shift the wheel gates as long

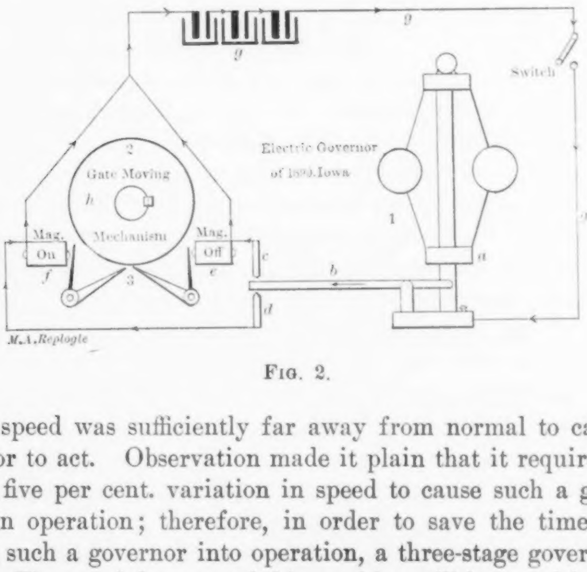

Fig. 2.

as the speed was sufficiently far away from normal to cause the governor to act. Observation made it plain that it required from two to five per cent. variation in speed to cause such a governor to begin operation; therefore, in order to save the time lost in getting such a governor into operation, a three-stage governor was made. The novel features of this machine will be noted in Fig. 2.

12. In the diagram a. is the speed governor, g.g. a battery with its circuit, and h. the gate-moving mechanism. With this combination a variation of 1 per cent. or less in speed was sufficient to close the circuit at c. or d. This in turn would energize magnets e. or f., causing them to put the gate-moving mechanism into motion. This governor was an improvement on the old type governors, as it utilized the most valuable time by beginning its

operations long before the old governors had varied enough for action. This machine required from one to five minutes to move the wheel gates from end to end. Any attempt to operate the gates in faster time resulted in over-running and hunting. I learned from this governor that the effects of gate opening always comes after the operation; and I concluded that a governor to be a success must "anticipate" the load variation. This seemed impossible, but I felt that there was room for improvement; therefore, I studied water powers carefully for several years. The result of this special study was a discharge gate governing apparatus in connection with the above electrical governor. Discharge gates

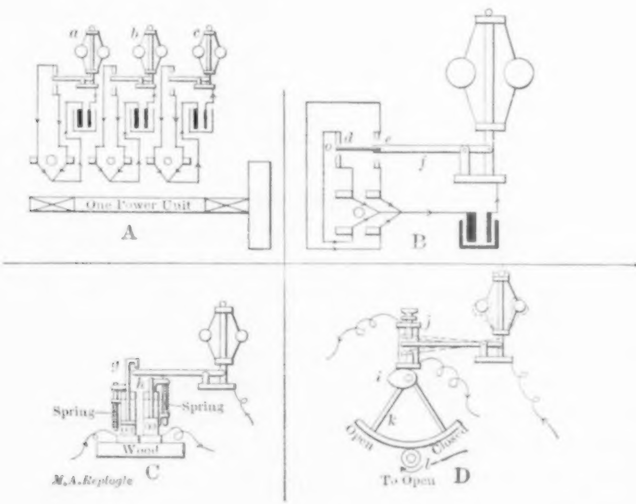

FIG. 3.

were not always practical; therefore, were abandoned after several fairly successful installations. The great point desired was faster gate movement, with no racing or hunting effects. I might add that discharge gate governing, in effect, was changing the head on the turbines at the same time that the quantity of water was changed. In other words, power was added in double the ratio of the ordinary methods, and the water-hammer effects were no greater. The racing effects were less because the gate effect followed its movements more promptly.

13. The illustration Fig. 3 represents about two years' experience. A. represents three speed governors, each one actuating

a gate regulator. The three turbines thus governed were all belted to one line shaft, making one-power unit. The principle to be brought out by this combination may be called cumulative or differential gate movement. Speed governor a. would cause its respective turbine gate to operate at one per cent. variation from normal. Speed governor b. would begin operating at two per cent. variation. Governor c. would cause its gate to operate at three per cent. variation; therefore, when a change of load would occur that would vary the speed as much as three per cent., all three governors would act, and the combined action was supposed to be fast enough to take care of the extreme changes of load in an electric railway power plant. The advantage of this scheme was a slow addition of power when the speed varied a little, and a rapid addition of power when a heavy change in load was made. This plan reduced the racing effect, even though a faster gate movement effect was used. Viewed from present-day knowledge there is nothing brilliant in this combination, but it taught me the value of differential gate movement, and as a matter of history it was the means of governing the first automatically regulated water-driven electric railway in the United States. The guarantee made to the owners of the water power was: "Better regulation than could be given by a man at the hand wheels." It was conceded that I had met the guarantee, and payment was made for the three governors.

14. In B. one speed governor inspires two gate regulators, or one double regulator by means of contacts o. and e. The flexible tongue d. allows lever f. to make a second contact with a greater variation in speed similar to action shown in A.

15. In C. contacts g. and h. are attached to the piston rods of dash pots or cataract cylinders. They were so arranged that when the speed governor would touch the contact the regulator would act and the pressure would force the piston through the oil as long as the speed varied from normal. A small spiral spring would cause the piston to return slowly to its former position after a change of load was made. The effect of this scheme was to cause the governor to act while the speed was going away from normal, and then by breaking the circuit the governor would stop acting, so that by the time the effect of the gate action was fully realized the speed would be returned to normal. This was the first device that I had ever used that would anticipate or make provision for the effects of the gate action coming after the gate movement. By a proper balancing of piston and spring a faster gate action could

be employed without racing effects. This was an important stage in the development, as it taught me that a machine could be made that would operate the gates only while the speed was varying from normal, and the contacts could be made to return to their original position, fast or slow, as the power storage effects of the plant would permit. Here was a crude relay governor, but I failed to ever find anyone else who could adjust it as satisfactorily as was desired. I was therefore compelled to devise a more positive and reliable means for accomplishing the same purpose. In D. the cam i. forms a support for contact-bracket j., which rises or falls by means of segment k., driven by gate-shaft l. With this device the governor would turn on water as long as the speed would drop, and would turn off water as long as the speed would rise. This permitted a more rapid gate movement without the racing effects being so marked. Here was a relay governor that offered an opportunity to operate wheel gates rapidly and in a great measure prevented overrunning; that is, in plants that embodied sufficient momentum in their revolving parts. But a new difficulty arose. A governor that could operate wheel gates from end to end in one-half minute must be four times as powerful as one to do the work in one minute, and it must be one hundred times as powerful if the work must be done in six seconds. Therefore, a new and more powerful series of governors had to be built. It was also learned that where power plants lacked in momentum the cam i. had to be so steep to prevent racing that the permanent drop in speed caused by it was too great for good automatic regulation. This was in a small measure remedied by making a differential cam to compensate as economically as possible with the variable effects of the gate openings. (Power is added fastest at the earliest stages of gate openings.) Therefore, the cam required greater drop on the start and decreasd rapidly as the gate opened. To overcome this large drop, I had to devise a governor that would permit the necessary drop in the first operation of the governor, while a secondary operation would permit a slow returning of the members to their original positions, and in doing so the governor, without change in speed, would operate on the wheel gates until the speed was correct. This governor was called a Relay Returning Governor, an anticipating machine. In making this governor it was necessary to perform so many of the functions mechanically that the electrical features were cut out of the governor entirely, necessitating another series of new governors. (There were other

considerations that made it seem best to drop the electric battery and its circuit.)

16. It can be truly said that all of the work up to this point was preliminary. I had merely learned the rudiments of successful water-wheel governing. These first principles had to be developed in order to know fully to what extent they could be employed under the many conditions of governing that are encountered in practice. The above constituted the first eight years of governor experience, and shows the development up to the time of preparing the paper for the Franklin Institute.

17. The past eight years have been devoted to the development of a machine that embodies the elements necessary to perform its functions in the proper ratios. I will state broadly that no two turbine installations govern alike, and the perfect governor must be susceptible of a greater range of adjustment than any other known machine. Its "reasoning" power must be prescribed for every power plant that it is placed in. That is, it must "anticipate" the rapidity of gate movement, as well as the point the gate should be moved to, for every change in load. Both of these judgments must be performed before the speed has begun to return to normal. It must then hold itself in suspense until gravity has fully charged the new water added with power, and this power is extracted by the turbine. If the speed returns to normal too fast or too slow, it must of its own volition shift the gates to a new point of opening. Here are four distinct acts of judgment that may take place in one change of load, and a smaller or larger change of load will require a change in all of the ratios. The above describes a simple change in the load of a turbine. In case a second or third change in load should occur before equilibrium for the first change has been established it is very apparent that the governor's actions must be further complicated. As stated before, a perfect governor must act with a precision not often found in the most studious man.

18. A few suggestions at this point will not be out of place. If a most perfect governor were attached to a turbine plant, it will not always assure perfect regulation, and the following are some of the defects found by the governor man:

TABLE 1.

Mechanical Defects.	*Hydraulic Defects.*
1. Unbalanced gates.	9. Long closed feed pipes.
2. Cramping or intefering in gates.	10. Contracted feed pipes.
3. Ponderous, unwieldy gates.	11. Short bends in feed pipes.
4. Lost motion in gates.	12. Slanting draft tubes.
5. Torsion in gate rigging.	13. Draft tubes too large or small.
6. Irregular effects of gatage.	14. Loss or gain of vacuum.
7. Lack in power storage.	
8. Obstructed intakes.	

19. Any one of the above defects will cause erratic action in the governor, and on account of some strange fatality the governor often gets full measure of credit for any or all of them. Space forbids discussion of Table 1.

20. *Table* 2 shows an aggregation of some of the more important conditions found in the power unit and some of the more important features a good governor must have, and by way of explanation will say that the word differential is intended to convey the idea of a constantly increasing or constantly diminishing motion, or effect. These motions or effects can be more accurately illustrated by the use of parabolic curves. (These statements are made from the standpoint of time.)

TABLE 2.

Power Unit Conditions Due to Inertia.	*Requirements of a Good Water-Wheel Governor.*
1. Speed variations are differential.	7. It must be extremely sensitive.
2. Effects of gravity are differential.	8. Must have differential gate action.
3. Speed governor effects are differential.	9. Must have differential temporary cut-outs.
4. All load changes are differential.	10. Must have differential return movement.
5. Gate effects are not immediate.	11. Must have permanent cut-outs.
6. Power storage effects are differential.	12. Should be adjustable to all ordinary conditions.
	13. Should have adjustable permanent drop.

21. Such a governor has been perfected. In preparing the way to the introduction of this machine I have omitted the details, and have referred only to the more important experiments. The ideas preserved were selected on account of their having special merit at the time of their employment.

22. The foundation of the new governor is a sphere and disk

transmission device. The practical limits of this device have been determined by eight years' experience. The physical make-up of the drive is the result of this experience, and has proven to be entirely satisfactory. In principle it is an application of the shifting belt and pulley. The only difference being that the pulley is shifted and the effect is a differential drive from a standstill to the fastest movement with full torque throughout its range.

23. Much experimenting has been done in the way of tripping or shifting devices during the past eight years with various degrees of success. During this time the new drive came repeatedly to the notice of Nathaniel Lombard, a most successful designer of

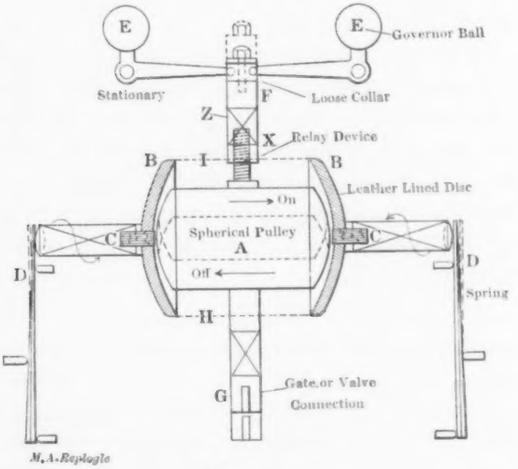

FIG. 4.

water-wheel governors. His fourteen years' experience in this work led him to believe that the ideal governor must be purely mechanical, and his search for a differential drive resulted in a combination; therefore the new governor is the result of our combined experience. Every member in its construction was the subject of a special conference, and it is the most perfect machine that our experience up to date has produced.

24. The governor in its simplest form is shown in Fig 4.

25. In the diagram A. is a spherical pulley with its shaft turned down and threaded as at X. B. and B. are oppositely revolving concave disks lined with leather. C. and C. are lignum vitæ pins flush with the leather. D. and D. are compression springs for

causing the necessary pressure between the disks and the sphere. (Please note that when the sphere is shifted from center, in the line of its axis, the springs are tightened automatically, causing increased traction, as the smaller diameters of the sphere engage the larger diameters of the disk.) E. and E. are governor balls so poised as to require the weight of A. to balance them at normal speed. F. is a loose collar to allow independent revolution of the balls E. E. G. is the point of connection between A. and the gates or valves of the motor to be governed. X. is the relay device, and is for the purpose of preventing racing. Also for the purpose of properly dividing the load in parallel units. Z. is a stationary spindle or connecting link between collar F. and the threaded shaft or pulley A. Z. is only stationary in reference to revolution, as it rises or falls with the variations of the governor balls.

26. The following is a description of the governor's action if the speed should drop by an addition of load: The lessening of the centrifugal effects on E. E. will allow A. to drop below the centers of disks B. B., which are constantly revolving in the directions shown in the diagram. As soon as A. falls below the disk centers it will begin to revolve slowly to the right, being the direction that will turn on power. While A. is turning to the right it shortens the distance to collar F. by means of the thread at X. This shortening causes A. to be pulled back to the disk centers, thereby cutting the governor out of action. It will be noticed that E. and E. have not shifted their position during the act of opening the valves. Therefore the speed is in reality lower after the new power is added than it was before the change in load. It is now clear that there is a continuous dropping in the speed while the valves are opening. In practice this permanent drop is enough to insure the correct division of load. It is also enough to permit of successful government where adequate power storage exists in the unit to be governed. In this governor there is no special provision for temporary relay. Such provision is unnecessary except where the momentum effects are small. (In the governor shown the permanent drop can be varied by the pitch of the thread used at X.) In ordinary practice it is about 2 per cent., and it can be less in steam turbine practice.

27. Before going further, it will be well to make some statement concerning the physical qualities of this governor. Experience has shown that we can secure 15 pounds pull per square inch of traction surface. The traction surface varies as the squares

of the diameters, with the odds in favor of the larger drives. In all of our experience the traction increases with the age of the machine. (Its action perfects its shape.) The leather in polishing the sphere causes greater traction. It requires very little power to operate it at normal speed. The increase in speed from shifting A. off center is not a constant increase, but a cumulative increase, such as is found necessary in accurate governing. The maximum speed of valves or gates can be fixed. In this drive or transmission the design can be varied so that the pulley A. can make two revolutions to one of the disks. The possible variation in Fig. 4 is a range from zero to the same speed of the disks.

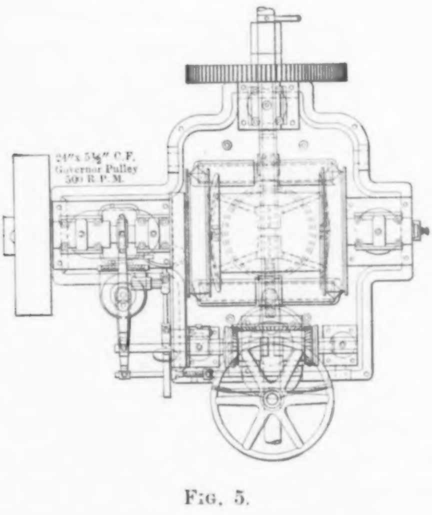

FIG. 5.

Springs D. D. can be so proportioned that greater torque effects will be produced at the higher speeds. That is, since the disks are concave, they will be forced back against the springs as the sphere is forced from its central or normal position.

28. Experiment has shown that the pressure necessary to force A. from the center is imperceptible until the concave surfaces begin to offer resistance. This tendency of the disks to press the pulley back to the center is in itself a valuable feature in a governor. (In effect it is temporary relay and is very reliable, needing no adjustment.)

29. Please note that the disks have a greater radius of curvature than the spherical pulley. If the radius of the disks should be

lessened, it will require a smaller movement of the pulley to bring about its fastest motion. Flatter disks will require a greater movement of A. to produce its fastest motion. It will be readily seen that this drive or transmission is susceptible of a great range in design, and it is therefore adaptable to a great number of mechanical transmissions. It might be called a perfect cone pulley drive in a most compact and economical form. It is a variable belt and pulley transmission, condensed.

30. The governor in this form is exceedingly sensitive to speed variations. Our experiments up to the present show that the

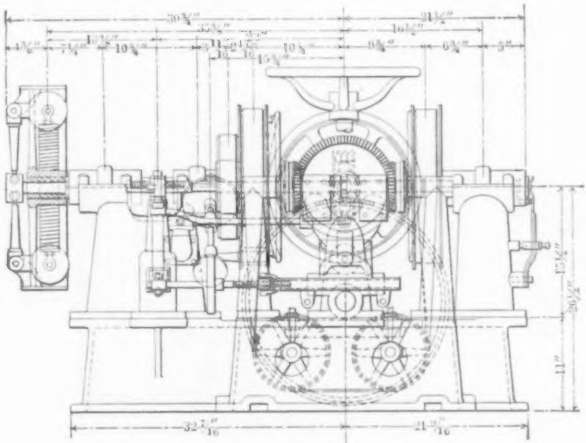

Fig. 6.

governor will operate in either direction, with variations in speed too small to be indicated by the ordinary tachometer.

31. Fig. 5 shows a plan view of a governor designed for the power house of the Sanitary District of Chicago. This governor is designed to develop 60,000 foot pounds of power in 8 seconds. The speed governor is in the driving pulley, and the disks are driven by belt. In this way we eliminate all gearing except the gate gears, and the small bevel gears necessary to move the sphere into action. This is a three-stage governor; that is, the power of the speed governor presses friction wheels into action, and they in turn force the power drive into action. The speed governor lever and the floating lever may be seen in the lower left-hand corner. In this governor the traction disks have a speed of 500 revolutions per minute. All fast running bearings are self-oiling.

This machine is noiseless in all of its actions, and requires a very much smaller proportion of power for governing purposes than has ever come to our notice before. In other words, instead of a continuous use for governing purposes of from 1 to 3 per cent. of the total power developed, this machine will require less than one-tenth of one per cent. This consideration alone is worthy of attention. The power required to keep the machine in motion is very small, and when in action the only further power required is the additional power necessary to operate the gates at whatever rapidity is required. There is an enormous saving over that class

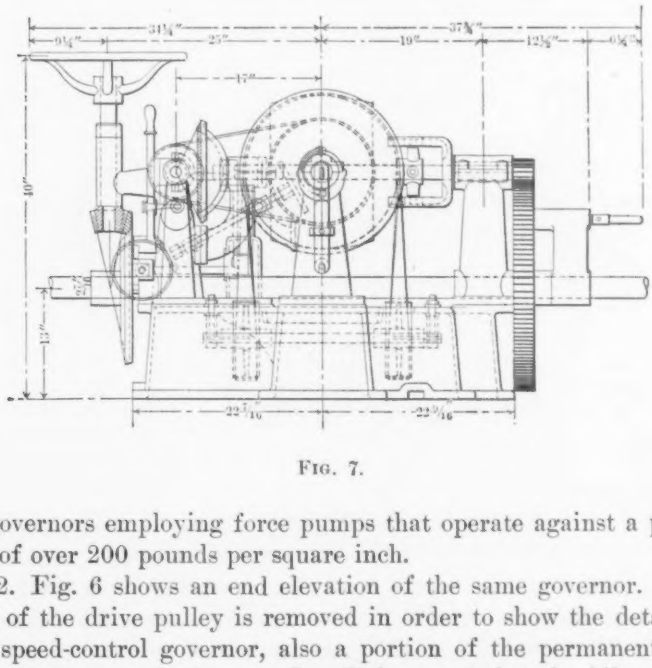

Fig. 7.

of governors employing force pumps that operate against a pressure of over 200 pounds per square inch.

32. Fig. 6 shows an end elevation of the same governor. The rim of the drive pulley is removed in order to show the detail of the speed-control governor, also a portion of the permanent cut-out arrangement is shown. It will be noted that the idlers used to guide the disk drive-belt are located in the base of the machine. The scroll gear and the hand wheel pinion are removed. The construction of the disk traction spring is also shown.

33. Fig. 7 shows a side elevation and shows fully the connection between the spherical pulley and the gate or valve shaft. The main shaft can operate at any velocity from standing to one revolution per second; therefore the governor can be geared to operate the gates in the smallest number of seconds that the conditions

will allow. The pin clutch at the right-hand end of the machine indicates the manner of throwing the governor positively out of action. The lever under the hand wheel is for the purpose of throwing the transmission into action, independent of the control governor. The slanting shaft in dotted lines furnishes the power to the temporary drop and time-element device. Its speed is varied by shifting the friction wheel nearer to, or further from the center of its driving disk.

34. Considering the many functions that must be performed, we believe this to be a most simple and perfect governor. The machine embodies the means for giving the necessary permanent drop as found desirable for the requirements of each power plant.

TABLE 3.

WATER-WHEEL GOVERNOR SERIES.

Type.	Capacity Foot Lbs.	Time in Seconds.	Pulley Diam. Face.	Speed R. P. M.	Height.	Base Dimensions.
L. R.-6	500	2	8″-2″	500	13″	19″x15″
L. R.-10	5,000	4	14″-3″	500	22″	2′-7″x2′-1″
L. R.-14.........	30,000	8	19″-4½″	500	2′-6″	3′-7″x3′-10″
L. R.-18.	100,000	12	24″-5½″	500	3′-3″	4′-7″x3′-8″

35. Table 3 shows a tabulation of capacity, etc., of the new "Lombard-Replogle" series of governors. It fully explains itself.

36. Fig. 8 shows a new invention used successfully where differential retarding or dash-pot effects are desired. This has been found very satisfactory during the past two years, and is a distinct member of a series of standard governors being manufactured. This apparatus embodies some novel ideas, therefore a short description will be given.

37. In Section X, A is a piston rod, having a piston B, with beveled edges or the corners removed. B is surrounded by the ring piston C. C is held in place by the lateral piston rods DD and the yoke E. E is centered by collars I and G by means of springs F and H. J and K are adjusting collars for the springs F and H. The purpose of this member is to permit a rapid movement of the piston without any great increase of pressure between its extremes. A careful examination will show that this device has a collapsible piston that operates equally well in either direction. Its first action is that of the ordinary dash-pot up to a predetermined pressure; after that it gives way with a very small increase

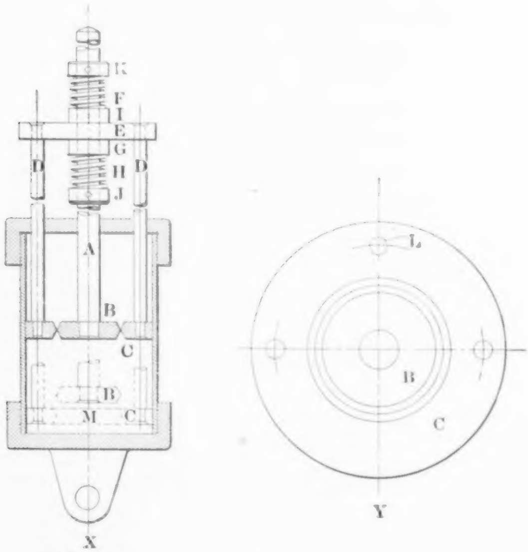

FIG. 8.

in pressure. This device has a large range of adjustment, and its function is to retard the action of the control element in a governor, so as to give gravity its time for action before the governor has done too much. By this cylinder the actions of the governor can be timed to the gravity and inertia effects of the power unit, making good speed regulation possible.

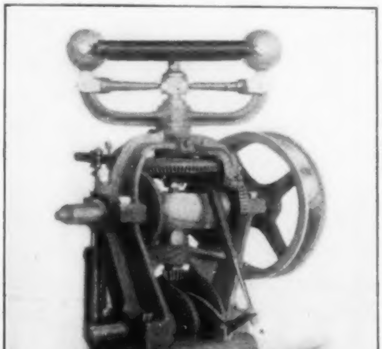

FIG. 9.

38. Section Y is a plan of the collapsible piston and shows the relative positions of B and C. L shows position of adjusting valve. M in Section X shows the possible position of B and C with a rapid movement of the piston. In the new type governors this member is replaced by a mechanical device, the great advantage of which is that it is purely mechanical; therefore more reliable than any hydraulic arrangement.

39. Fig. 9 is made from a photograph of a machine that is now governing a steam turbine in Texas. It is the first of this type, and is therefore subject to changes as far as general appearance is

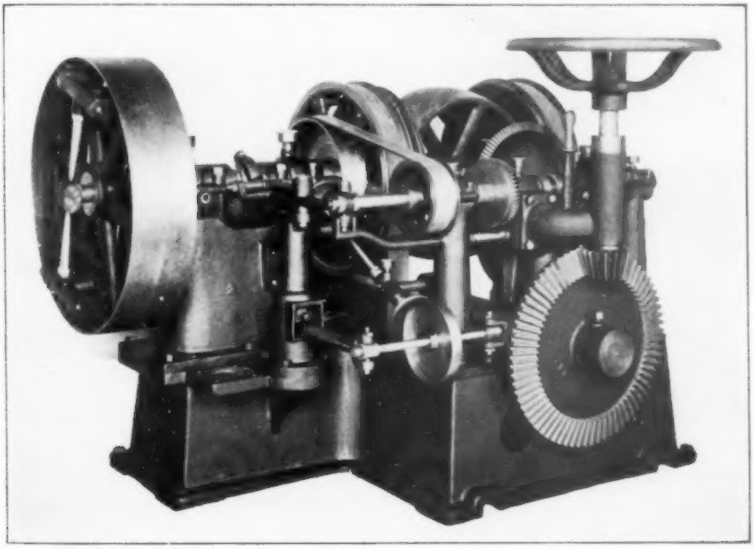

FIG. 10.

concerned. Its present form was for the purpose of demonstrating the principles involved. We have not yet received tabulated data concerning its performance, but have been informed that it meets the requirements.

40. In conclusion will say that all good engineers who have given the subject careful thought agree that the ideal governor should be simple and easily managed.

It should be carefully and compactly designed.

It should have a powerful and exceedingly sensitive control governor.

It should embody a powerful and reliable gate-moving element, whose every movement is gradual or of a differential character.

It should be limited in its velocity of gate movement to the hydraulic and mechanical conditions of the power unit.

It should have enough temporary and differential drop to make it adjustable to the present day power unit conditions.

It should have an adjustable permanent drop, so that this factor can be reduced to a minimum in the plant governed.

It should be provided with means for temporarily changing the speed for synchronizing purposes, and this feature should be controlled from the operating board of the modern power plant.

It should be built as accurately as present-day methods will permit.

All of its important bearings should be self oiling.

It should be noiseless in all of its actions.

It should require a minimum amount of power for the performance of its duties, and when it is not laboring but is waiting for inspiration, the energy required should be reduced to a mere driving of empty belts.

THE IDEAL GOVERNOR SHOULD BE MECHANICAL IN ITS EVERY DETAIL.

DISCUSSION.

Mr. Fred'k W. Salmon.—I should like to ask the author for his views and any data relating to the operating of alternating current generators driven by water turbines in parallel with steam engines, if any special difficulties have been experienced? What they were? The cause of them? How they were overcome and to what extent, and what precautions should be taken to prevent such difficulties and others likely to arise in such work?

*Mr. Mark A. Replogle.**—I have had some experience in governing water turbines in connection with steam engines and especially reciprocating engines. The oscillatory motion of a crank engine is somewhat steadied by running it in multiple with a water wheel. This, of course, is due to combining the weights or momentum of the revolving parts. The main difficulties are usually found in the characteristics of the governors. In order to insure perfect operation, the characteristics of both the steam governor and the water governor must be the same. That is, the permanent drop in speeds of both governors, as well as the variation in curves

* Author's closure, under the Rules.

due to the load, should be alike. To get two such governors properly synchronized requires some experimentation. There are no insurmountable difficulties, and on account of the water governor being susceptible to all kinds of adjustment, it can usually be adapted to the steam governor. The same difficulties are found in steam engines of different designs. Often a very material change must be made in the governors of large steam engines in order to make their characteristics nearly enough alike to insure equal division of load. Referring to the query "what precautions should be taken to prevent such difficulties?" I will suggest that there should be a conference of competent engineers for the purpose of establishing standard characteristics so that the designing engineer can specify them when calling for propositions. The problem of governing alternating units in parallel is of such importance that the purchasers of governors, whether for steam or water powers, can well afford to purchase intelligently, especially when they hope to run various units in parallel.

No. 1111.*

THE REGULATION OF HIGH-PRESSURE WATER WHEELS FOR POWER-TRANSMISSION PLANTS.

BY GEO. J. HENRY, JR., SAN FRANCISCO, CAL.

(Member of the Society.)

1. When the modern engineer is presented with a problem of safely, economically and properly governing a power-transmission plant, there are a number of points to receive his exhaustive consideration, aside from the construction of the governor itself.

2. A modern high-pressure water plant is usually exposed to the danger of a break in the pressure pipe. Such a pressure pipe frequently carries a very large volume of water at anywhere from 200 to 1,000 pounds per square inch pressure; and a break in this line in many instances would mean the complete wrecking of the power house. It is, therefore, extremely necessary that the security of this pipe line be guarded in every way; and the old method of regulating turbine wheels by throttling the water is one that should not be adopted without installing suitable protecting devices, and even then, only when it is the only available method. Of course in many instances, particularly where a large storage reservoir is available at the inlet of the pipe or at the outlet of the flume, or where the peak load is likely to exceed the normal capacity of the flume (which may be a long one and very expensive), or where it will exceed the normal flow of the stream, it becomes advisable to save all the water possible; and this can only be done by proportioning the water flow to the station load requirements. Some form of regulating nozzle is of course the best way of accomplishing this result, but it does not by any means follow that the regulation of the speed should be coincident with the regulation of the water flow, as will be pointed out below. It is very necessary, in modern power-transmission plants, to maintain the speed within

* Presented at the Chattanooga meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

very close limits, and $1\frac{1}{2}$ to $2\frac{1}{2}$ per cent. is readily obtainable with well-designed and constructed apparatus. It is not customary to attempt to maintain voltages at the end of a transmission line by varying the speed, and therefore the voltage at the generator, but modern practice invariably requires the maintenance of a constant speed and voltage. Fluctuations in voltage are then taken care of by varying the generator field or by accessory apparatus, as, for instance, the Tirrell regulator.

3. In many cases it is not permissible to vary the flow of water in the pipe line, the requirements of the irrigation district below the plant frequently being such (and the courts have sustained this) that the full flow of the stream must be permitted to pass through the plant at all times. It therefore becomes impossible to regulate the speed of a plant of this kind by varying the water flow, unless an accessory spill-way or pipe line feeds into the same water discharge channel that water which overflows the pipe inlet.

4. It is obvious that if the governor which is to be used to move the gates, deflecting nozzles or other means of regulation, is not provided with a "relay" and a "relay returning device," that the speed is very apt to "hunt," causing dangerous fluctuation in the entire system, if not absolutely preventing successful operation. It is also obvious that the rapidity of the governor operation will determine for any plant with a given amount of stored energy in the rotating parts, the variation in speed which occurs before the governor properly checks and corrects the rise or fall in the speed. On the other hand, every mechanical governor requires a certain change in its speed before it tends to correct the variations. This is due to the lap of the valves, the friction of the parts, etc. Again, in a plant having a large amount of stored energy, a given load fluctuation will produce a slower speed fluctuation. Hence the governor will get into operation more slowly, and although oscillation or hunting is still likely to occur, its periodicity will be longer.

5. On the whole, it may be stated that while the greatest permissible rapidity of governor control is highly desirable, the addition of fly wheels with the necessarily increased windage and bearing friction are not in ordinary cases desirable. Governors of excellent design and machine construction are to be had from several manufacturers, so that the problem which confronts the engineer of to-day is usually that of properly adapting the apparatus that is readily obtainable to the conditions to be met with in

any individual plant. The general practice among turbine builders, as stated above, is to control the speed of their turbines by cylinder or wicket gates, either device aiming to limit the orifice of discharge without varying the spouting velocity, and therefore maintain high fractional load efficiencies, although neither of these devices do this perfectly. They both are open to the defect of causing a variation in the velocity of flow in the pipe line, and therefore a corresponding water ram every time the governor checks the velocity. There is not, ordinarily, any trouble experienced, except in cases of extremely low head, due to the governor opening the gates too quickly and the water not getting up to spouting velocity quickly enough; but there is frequently very great damage caused by the governor shutting the gates so fast that the pipe or the turbine case is ruptured by the resulting water ram. In order to guard against this, it is customary to install safety valves, which usually are out of commission, owing to their requiring too great an increase in pressure in order to actuate them, or due to their freezing or becoming stuffed up with leaves, sticks, etc., carried by the water. Where there are a number of turbines fed by the same penstock, the water ram, due to a single machine closing, is not so serious as where there are but one or two on the same line; but the resulting change in spouting velocity, due to the monetary water ram, is then an objectionable feature, tending to cause a further increase of speed variation. With the Pelton type of water wheel, where a stream of water issues at full spouting velocity from a nozzle and enters the double-curved surface of a suitably designed bucket, its velocity is almost entirely taken up and the water caused to discharge from the sides of the water-wheel buckets at zero velocity. This type of wheel readily lends itself to a construction utilizing a deflection of the stream of water from the buckets in order to reduce the load that the wheel will carry at any instant of time. Such deflection can readily be accomplished without interfering with the spouting velocity of the water from the nozzles, by merely diverting the stream off or on to the buckets. This can, of course, be done by pivoting the nozzles, or the nozzles may be made rigid and a stream deflector introduced into the jet in front of the nozzle tip.

6. Fig. 1 is a view of a Pelton wheel in operation when running at the correct speed, and shows one bucket just entering the stream of water; another bucket advanced to a mid-position and

receiving the full impact of the jet, and the third bucket receiving the remaining portion of the jet, which has been cut off and is now flowing into the second bucket. The discharge from the sides of the buckets is clearly shown, and the fourth bucket, although not receiving water, is clearly shown to be discharging it, it having received its section of the stream which is still flowing over its interior surface and discharging from the outer edge. This photo

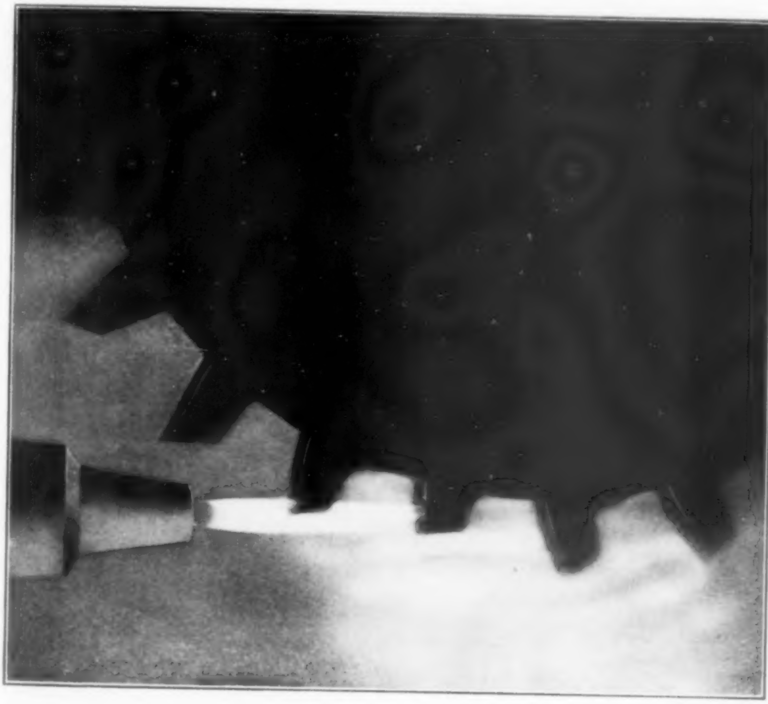

FIG. 1.

was taken with a special apparatus, the wheel being illuminated by an arc lamp in front of which is rotated a shutter exposing a ray of light at every instant that a bucket passes a given spot. (The apparatus was fully described before the Pacific Coast Transmission Association at San Rafael, June 16, 1903.)

7. Fig. 2 is a model showing a pivoted deflecting nozzle, which is arranged for moving up and down by the automatic governor which is run by suitable belt from the wheel shaft.

8. Fig. 3 shows a triple nozzle for application to a Pelton wheel,

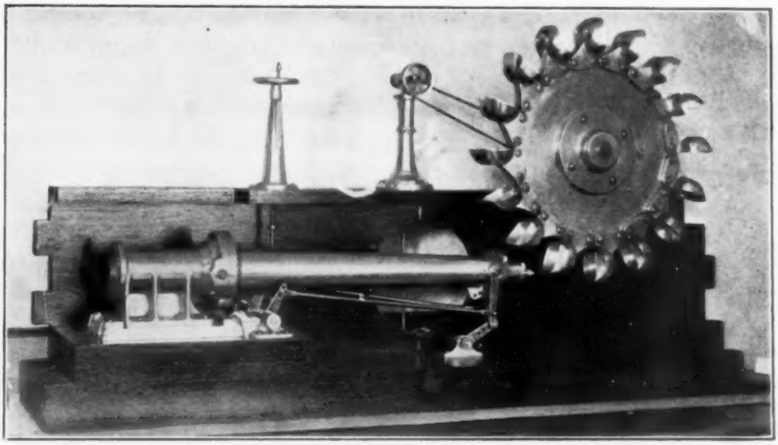

FIG. 2.

each outlet of which is fitted with a stream cut-off. These cut-offs vary the size of the opening, and therefore, while not varying the spouting velocity of the water issuing from them, vary its cross-sectional area, and consequently cause more or less water ram in the pipe line, but economize in water. The objectionable feature of the deflecting nozzle is its wastefulness when operating the plant at less than full load, and the objection to the cut-off is that

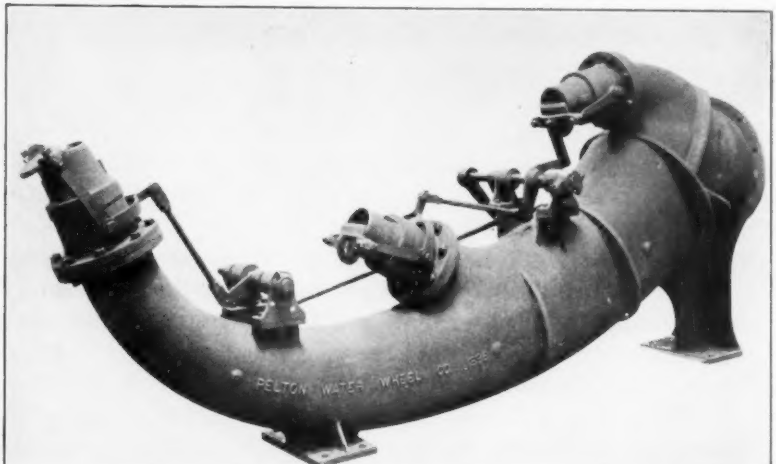

FIG. 3.

it exposes the pipe line to constantly recurring shock exactly the same as a turbine gate. In order to economize with water and secure accurate governing, it has been the custom in recent years to install a combination needle and deflecting nozzle. The needle nozzle is one in which a suitable curved central core is provided to direct the stream of water issuing from the nozzle, the nozzle tip being also curved to properly direct the stream over this surface. Such a nozzle is shown in Fig. 4, the needle projecting beyond the surface of the nozzle tip; and in Fig. 5 the needle is

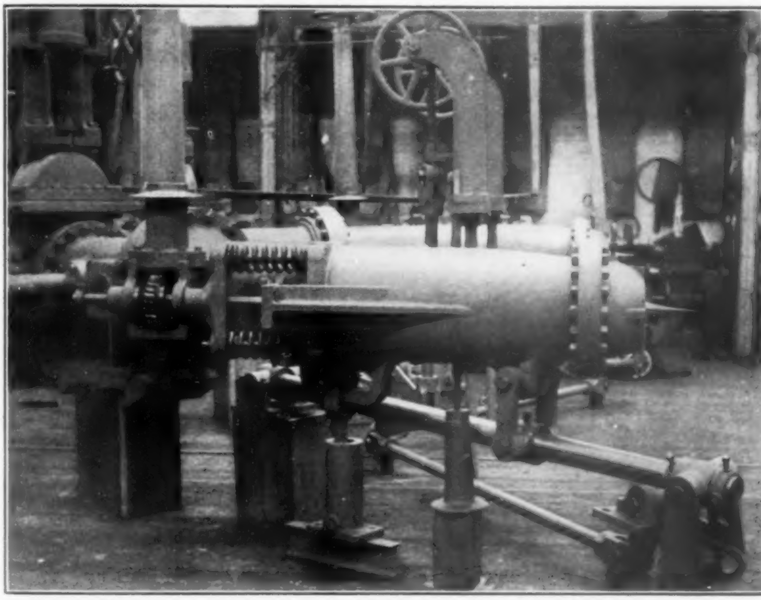

FIG. 4.

shown through the transparent stream of water issuing from the nozzle. This particular nozzle is operating under 390′ head and has a capacity of 1,500 horse-power. It is obvious that if we regulate the position of the issuing stream by a suitable governor acting on a deflecting nozzle or a stream deflector, we may actuate this governor as quickly as we choose without interfering with the safety of our pipe; and then if we vary the position of the needle in the nozzle we can vary the cross-sectional area of the issuing stream, and thus save water; this, of course, being done at such a speed as not to cause a dangerous increase in the running pres-

sure. In following this course we must, however, avoid closing the needle to such a point that a sudden load coming on, and the governor raising the nozzle into the wheel, there will not be sufficient water to generate the required power. In practice, the combination needle and deflecting nozzle is used by setting the needle to a point corresponding with the peak, which is likely to occur during each hour, and then allowing the governor to take care of the speed variations up to this peak. By this course some water is naturally wasted, but there is also a very considerable saving effected, and, except in the largest plants, it would hardly be worth while to install special automatic apparatus for effecting

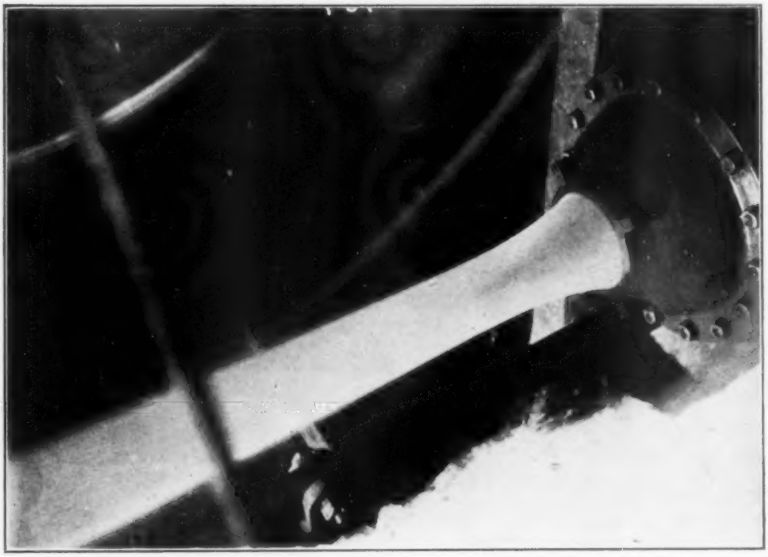

FIG. 5.

a further saving. To handle these large needles and also these large deflecting nozzles quickly requires a very considerable amount of power. Two such nozzles as shown in Fig. 5 require about 12,000 foot pounds, and when operating under 890′ effective head are capable of developing upward of 10,000 horse-power from the two water wheels which are mounted on a single shaft. The type of unit on which this is used is clearly shown in Fig. 6, the governor being arranged in the center between the two gate valves and controlling both of the nozzles. The floor stands for oper-

ating and their indicators for showing the positions of the needles are shown on each side.

9. In order to further reduce the power required to handle these large nozzles, counterbalancing cylinders may be introduced under the nozzles, supplied by pressure either from the governor oil-pump system or from the main pressure pipe line, as shown in Fig. 7. It is, of course, advisable to keep all of the water pressure parts and governor operating parts, except the delicate mechanism, below the floor line, and it is customary to cover the pit in which they are located with sheet steel grilled floor plates. The curve, Fig. 8, will give some idea of the amount of water saved

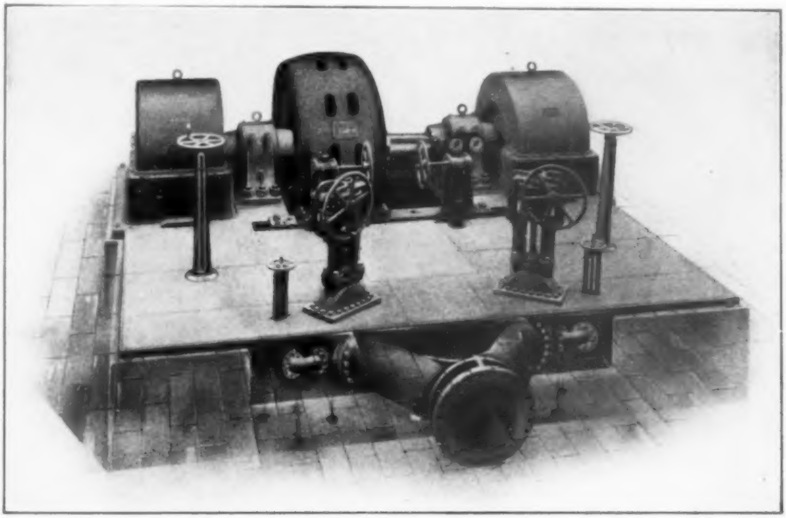

Fig. 6.

by using the combination needle and deflecting nozzle. The amount of water required to handle the plant, if we had an absolutely perfect gate operating mechanism—*i.e.*, one in which the water quantity would be directly proportional to the kilowatts delivered from the generator—would be shown as the area within the curve A. The amount of water required by the needle deflecting nozzle where the needle is set to the peak that will occur within any hour is shown by the curve B; and the amount of water that would be required if we used a straight ordinary deflecting nozzle would be that included in the entire parallelogram C. Of course the difference between B and C may be turned into a

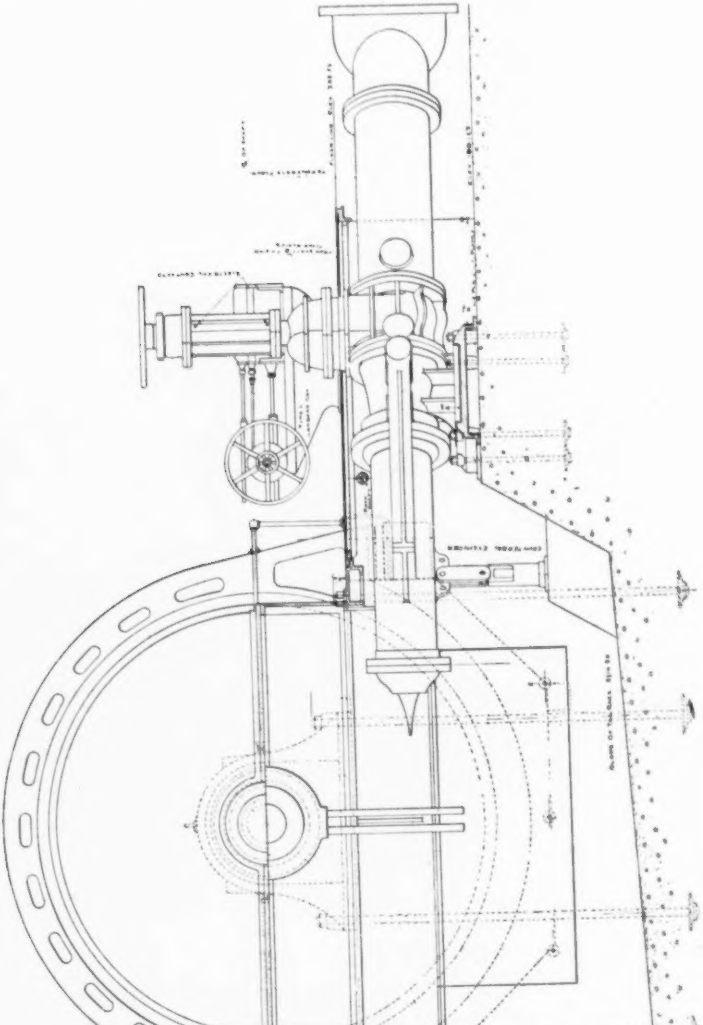

FIG. 7.

reservoir and result in the Power Company selling a greater output from their plant, using the same original water supply. Such an approximately increased output curve is shown by the line D. In order to save the water quantity between A and B it has heretofore been necessary for us to sacrifice the safety of our pipe by using a governor acting directly on the needle nozzle; and in order to compensate for this additional risk as far as possible, it is advisable to use safety water relief valves. Such relief valves, if of any value, will, of course, permit the escape of some water from

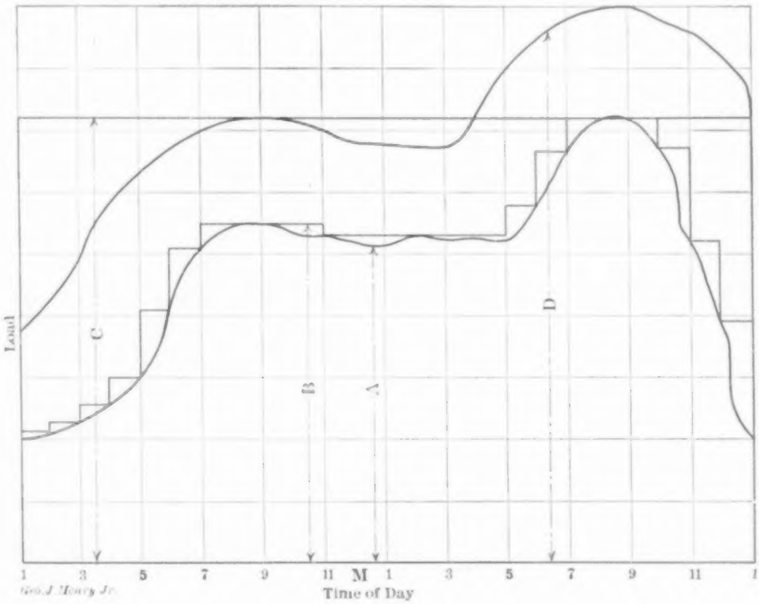

FIG. 8.

the pipe line whenever the governor closes the gate quickly, and will accomplish the same result as setting the needle of the deflecting nozzle by hand at more frequent intervals, thus drawing the curve B closer and closer to the curve A.

10. In the average installation, even although water economy is important, it would hardly pay to introduce expensive or complicated devices for the purpose of taking care of this slight saving between curves A and B. On the other hand, power plants are becoming larger, and water power more expensive to develop, making the value of water greater, and making the units of much

larger capacity, thus allowing the introduction of comparatively few automatic devices to save a quantity of water corresponding with a much greater horse-power than was possible a few years ago.

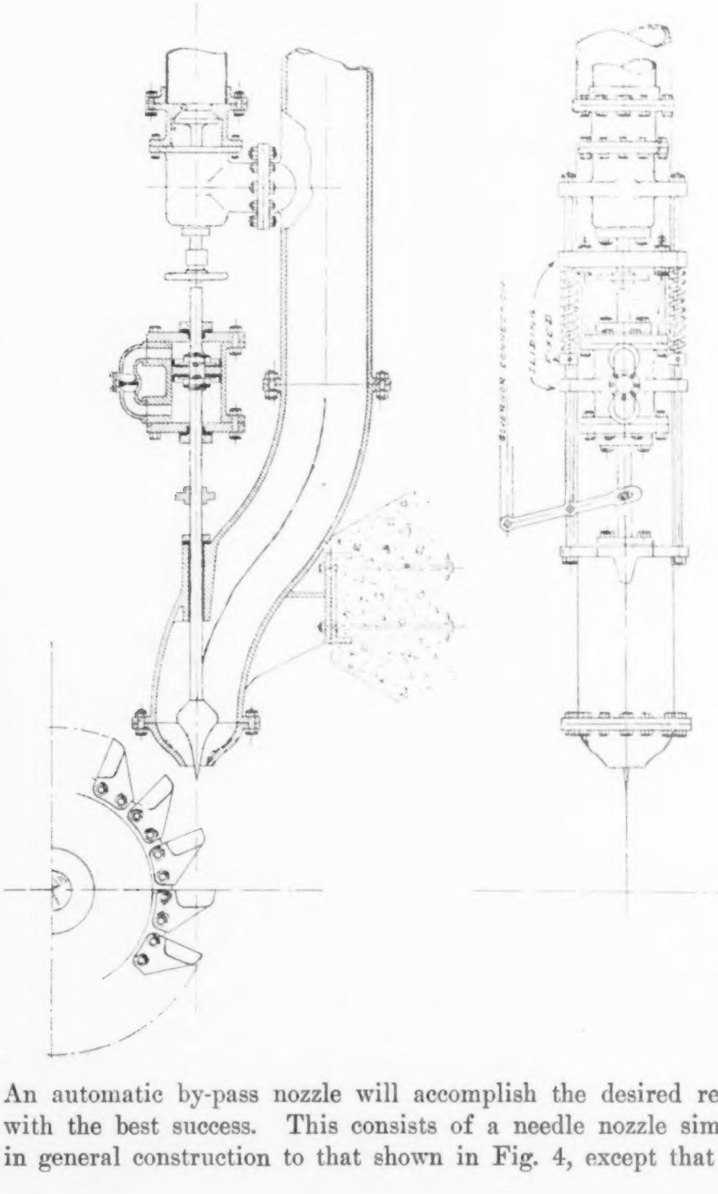

Fig. 9.

An automatic by-pass nozzle will accomplish the desired result with the best success. This consists of a needle nozzle similar in general construction to that shown in Fig. 4, except that the

needle is pulled back from or advanced into the tip by the governor, thus allowing a greater or less flow of water on to the buckets of the water wheel; and coincident with the needle's action a by-pass is operated admitting a discharge of water from the nozzle body when the effective stream is reduced. This discharge of water is only momentary, the by-pass valve immediately starting to slowly close, its rate of closing being dependent upon the length of the pipe line and the permissible rise of pressure above normal. Such a device is shown in Fig. 9. It will be noticed that the successful operation of this device does not in any way require or depend upon the raise in pressure, due to the water ram, but is entirely independent of this. Moreover, the by-pass for the dash-plot cylinder is arranged with a double valve, so that when an increased load comes on the water wheel the needle is quickly pulled back, the oil in the by-pass cylinder then being allowed to return to the front compartment with much greater rapidity. This device, therefore, secures for us the best possible regulation by quickly varying the effective stream's cross-sectional area, which is attained with the greatest degree of safety to the pipe line by preventing water ram. It can also be handled by a comparatively light governor, as the parts can be well balanced and require a very small amount of power to handle them. This by-pass nozzle can, of course, be built in a number of ways, but Fig. 9 shows one of the best and simplest constructions for it.

11. It is obvious that if we eliminate the dash-pot cylinder and properly construct the curves of approach to the by-pass outlet, we can then vary the cross-section of our effective stream without interfering with the velocity of flow in the pipe line, permitting whatever water may be cut down from the effective stream to discharge through the by-pass outlet, thus securing, if necessary, a constant rate of discharge through the nozzle, and at the same time obtain accurate regulation on the water wheel. In practice, however, if it is desired to attain this object, deflecting nozzles will probably be found more satisfactory. They may be readily counterbalanced, either by weights or in large installations by hydraulic or oil pressure through suitable cylinders arranged immediately under the nozzles. They do not in any way interfere with the flow of water in the supply pipe, nor can any damage that would in ordinary practice occur to them be likely to cause any interference with this. Where the deflecting nozzle is used for speed regulation we may rest assured that we are obtaining the maxi-

mum safety of our pipe line. The connections for operating such a deflecting nozzle as installed in most of the best plants are shown in Fig. 4, where tension rods, operating through suitable bell cranks and universal joints, raise and lower the two deflecting nozzles operating on the two wheels of a single unit (of the type shown in Fig. 6), such control being effected by a single governor. In the case of a 10,000 horse-power unit operating under 890′ head, about 12,000 foot pounds is required to properly move these nozzles; and in order to secure good regulation, this work has to be performed in from two to two and one-half seconds, the generators carrying a railway and gold-dredger load. However well this apparatus may be constructed, there is a sufficient amount of lost motion to prevent their accurately arriving at the position demanded by the governor; and therefore, when the parts have once come to rest, the wheel speed may be, at times, sufficiently removed from the normal speed to cause the governor to again shift them. This lost motion, therefore, has the effect of causing the governor to "hunt," although within narrow limits (in practice within the 2 per cent. mentioned above). Again, to properly construct and care for these heavy rock shafts, it is quite expensive, and their lubrication is quite an important feature.

12. In order to correct these defects, particularly in large plants having a number of units, the construction shown in Fig. 10 is advised. In this case each nozzle is entirely controlled by a pressure cylinder located immediately over or under it, and, if desired, a separate cylinder may be used for counterbalancing the nozzle weight, or the operating cylinder may be made of the differential type. This construction permits the removal of all the expensive connections heretofore located between the governor and the nozzles, thus eliminating the difficulties experienced, due to lost motion and the large expense involved. The governor is of the usual type and admits pressure fluid, usually a special mixture of oil, into one side or the other of the nozzle operating cylinder, thus raising or lowering the nozzle without the intervention of other connecting means than a pair of links. The governor may be located above the floor line and the piping connections carried to the cylinder under the nozzles. As the nozzle moves to take up its new position, it actuates through a small connecting rod a piston located in a displacement cylinder, which displaces a small amount of oil, and thus resets the piston valve which controls the flow to the operating cylinder, this piston valve being operated on the other side

by the fly-balls actuating a small controlling piston valve. In order to properly synchronize a number of machines a small, reversible motor may be located on the governor connections and

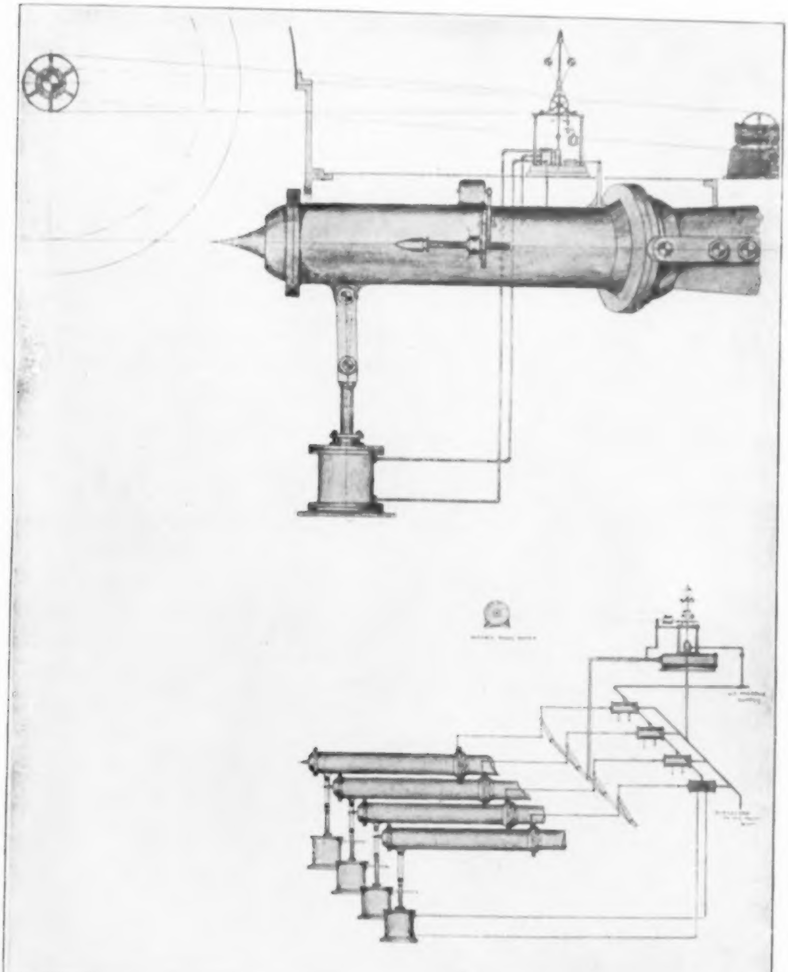

FIG. 10.

actuated from push buttons on the switchboard, this motor lengthening or shortening the pilot valve stem, which is actuated by the fly-balls. This will enable the operator to start up the wheel from

the switchboard by gradually raising the effective stream into the wheel by shifting the valve stem, and thus permitting oil to flow to the outside of the operating cylinder. In all modern plants a number of units are operated in synchronism, feeding into a single set of bus-bars. Therefore all of the units run at absolutely the same speed, regardless of the position of the nozzles, and unless the nozzles are all about of the same relative position one of the units carries a greater amount of load; or unless the governors are adjusted exactly right, some of the nozzles are likely to be entirely deflected and their machines running as motors. It is therefore advised that there be arranged a single governor driven by a small synchronous motor, which in Fig. 10 is shown belted to the governor for the sake of clearness. This should in practice, however, be direct-connected and mounted on the governor table, the governor being provided with a suitable set of valves and accessory operating cylinder and relay and relay-returning devices. Any variation in speed that occurs on the main unit instantly causes a corresponding speed change in the governor balls and causes the governor to shift its piston an amount corresponding with the change which will be required in the setting of all the nozzles in order to properly adjust them to the new load requirements. This governor piston shifts the long vertical lever shown in the figure, thus operating the four pilot valves which admit pressure fluid to one side or the other of the operating cylinders located under each nozzle. All of the nozzles being controlled by this single governor will then move, although it is not necessary that they should move at the same speed or that they should require the same amount of work to move them. Each nozzle as it moves will gradually shift the pilot valve piston back into its original position, through the action of the floating levers, until the ports of the nozzles' operating cylinders are again closed. All the nozzles will therefore take up a position exactly corresponding with each other, and their position may be indicated above the floor line by the position of the long, vertical lever connected with governor piston.

13. It will be observed that an adjustment can readily be provided on the rods connecting the floating levers with the pilot valves to adjust the pilot valve setting with respect to its own nozzle, thus enabling the station operator to unequally distribute the load on the different units, or in starting to properly synchronize them.

14. If desired, this same apparatus may be applied to stream

deflectors, the operating cylinders shown in the figure under the nozzles being then arranged to shift the stream deflectors. Such an automatic counterbalanced stream deflector is shown in Fig. 11, the interior surface of the stream deflector being curved so as to secure a reaction equal to the pressure, thus enabling this device to be moved into or out of a very large high-pressure stream with an extremely small effort; and the operating cylinders may be arranged to move the needles in or out of the nozzles, or they may be applied to the by-pass nozzles mentioned above; in any event,

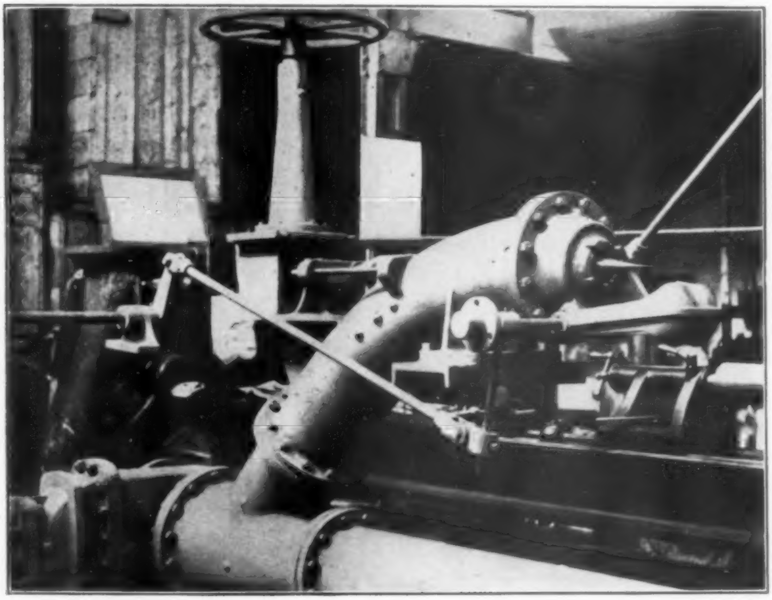

FIG. 11.

enabling the entire plant to be operated from a single small governor, located at the most convenient point in the power house, thus doing away with a large amount of expensive apparatus and, what is more, simplifying the design throughout and securing better regulation. The common rock shaft which shifts the floating levers should, of course, be made of such size as to practically eliminate torsion, but to properly handle the largest nozzles will require but a very few foot pounds of work as against many thousands heretofore; so that this is a point easy of attainment.

15. It will, of course, be noticed that the use of the needle and

deflecting nozzle does not provide a means of suitably governing turbine plants, and the present state of the art is such that the use of safety valves is almost absolutely necessary to properly protect the pipe line when governing them on railway and other rapidly fluctuating loads. Such a valve is shown in Fig. 12. In this design a double-beat safety valve is provided on the main pipe

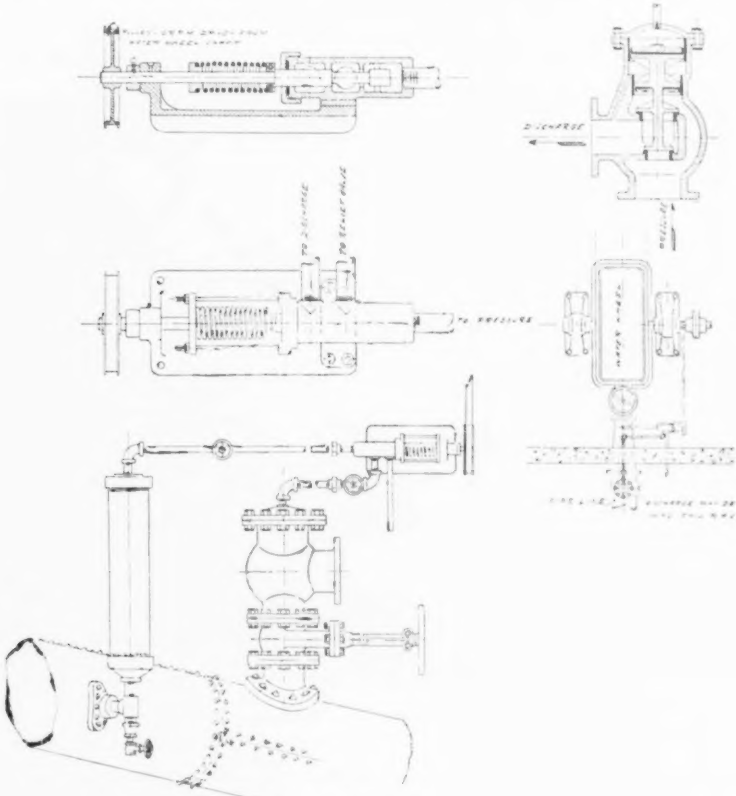

FIG. 12.

line, in which valve chamber both valves are held securely on their seats by fluid pressure entering a cylinder and exerting a heavy pressure on the piston, which is directly connected with them. The removal or reduction of this cylinder pressure (shown as the upper chamber in the valve) will serve to release with greater or less rapidity and to a greater or less degree the double valve from its seats, thus allowing a suitable discharge of water from the main

pressure line to relieve the water ram. It is therefore necessary for us to first reduce and then slowly restore the pressure in the top cylinder (Fig. 12), if we desire to relieve the water pressure in the main pipe line. In order to accomplish this a connection is made from the main pressure line to a rotating pilot valve. This pilot valve is shown in section and plan, Fig. 12, the central side outlet being connected to the cylinder in the top of the discharge valve, Fig. 10, and all being arranged in connection with the water-wheel apparatus as shown in the lower diagram.

16. It will be observed that water pressure against the end of the piston in the pilot valve tends to compress the adjustable spiral spring, and the piston takes up an intermediate position just sufficient to maintain pressure on the piston of the double-beat safety valve. If now an increase in pressure occurs in the main line the pilot-valve piston is driven forward, closing the pressure connections from the relief valve and allowing the relief valve to operate through the pilot-valve port to the discharge pipe, therefore instantly relieving the pressure which maintains the double-beat valve in its closed position, allowing the valve to open and free the pipe line of its excess pressure. As soon as this excess pressure is properly relieved and normal pressure restored by the escape of the water through the double-beat valve, the pilot-valve piston returns to its original position of cutting off the discharge from the relief valve, and restoring the connection between the relief-valve cylinder and the main pipe, thus closing the relief valve.

17. It will be seen that the action of this valve is very positive, and yet its operation occurs within extremely narrow limits of pressure variation, if the spiral spring is suitably adjusted for this purpose. The piston is continually rotated from the water-wheel shaft or other source of motion, in order that it shall more quickly respond to pressure variations and not by any possibility stick in the packing gland.

18. A feature of advantage in this construction is that a very small quantity of pressure fluid is used for operating the safety valve—not more at any time than the volume of the cylinder. Therefore, a very small velocity occurs in the pipe leading from the main pipe line to the pilot valve, and there is consequently a very much less chance of its becoming plugged up with leaves, sticks, etc., and a settling chamber may be introduced in this pipe line with a suitable blow-off valve, if the water is likely to carry ma-

terial which would in any way interfere with its operation. As a further precaution, the connection to the pilot valve may be taken from the side of the pipe instead of the top or the bottom, thus avoiding material that would float or that would sink in the water column traversing the main pipe. A valve of this type with its pilot-controlling mechanism is of course more expensive than the ordinary spring-actuated safety valve, but it can be depended upon to give very much more accurate regulation, and is a much better protection to a pipe line than any valve of the spring-actuated type.

DISCUSSION.

Mr. Frederick W. Salmon.—Mr. President and Gentlemen: I should like to inquire about the operating of alternating current generators driven by water turbines, or Pelton wheels (impact or reaction water wheels), in parallel with steam engines. Have any special difficulties been experienced? If so, what are they, what are the causes, and how are they overcome? What precautions should be taken to prevent such difficulties and others likely to arise in such work?

*The Author.**—The subject of the paralleling of alternating current generators has been very ably discussed at a number of meetings of the American Institute of Electrical Engineers, where, I believe, very complete data on this subject will be found. I do not think any greater difficulty should be experienced running alternating current generators driven by water power and by steam engines in parallel than if both are driven by steam engines. As far as satisfactory electrical regulation is concerned, I believe this can be depended upon more satisfactorily from water-driven generators than from those driven by reciprocating steam engines. It is the practice in several plants running water-driven generators in the mountains, transmitting to sites where the power may be supplemented at times by current from steam-driven generators, to do the speed regulating at one of the other power houses; that is, adjusting the governor of one or the other equipments to lag behind slightly. Suppose, for instance, we adjust our governors so that the steam engine will be entirely shut off by its governor before the governor in the hydraulic power house would begin to close off the water, there being allowed a very slight difference in speed, to which the governors were

* Author's closure, under the Rules.

adjusted for this purpose. The result would be that our hydraulic equipment would be operating at full output continuously, and the necessary amount of overload provided by the steam equipment, thus making the best use of the water and conserving the fuel at the steam plant. I believe this is in line with the best practice, and know of no instance where trouble has resulted from it. Where a number of hydraulic or steam plants are being run in parallel, the same course may be pursued, conserving the water or fuel at those points where it will have the greatest value.

No. 1112.*

SPEED REGULATION OF WATER-POWER PLANTS.

BY JOHN STURGESS, TROY, N. Y.

(Member of the Society.)

1. In this paper an attempt will be made to present certain aspects of the problem of regulating water-power plants, which are usually passed over with little consideration. These aspects relate to certain fundamental elements of design in the water wheel itself, and to the characteristics of governor action as obtained in the best practice of the day.

2. It has seemed to the author that much inaccuracy of conception exists on this subject, even among those more or less in daily touch with water-power development. This is no doubt largely due to the evolution of water-power work, and the tendency to accept as final certain features of the plant which may have fully answered requirements a decade ago, but which are in need of study and revision to meet present-day needs.

3. The commercial importance of satisfactorily regulating the speed of such plants is now fairly well recognized, though it is surprising how often, even at the present day, the whole question of governing is treated as a matter of minor importance when the designs of the plant are in course of preparation. It has frequently occurred within the author's experience that the generators and water wheels of a plant will be completed and installed before attention is turned to the governors. The consequence is that the latter unavoidably wears the aspect of a patch, is needlessly complicated and inconvenient, and is unnecessarily expensive, both to construct and maintain. It is not, however, our purpose to discuss this question now.

4. In regulating a water wheel it is not sufficient to confine

* Presented at the Chattanooga Meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

our attention to the governor only, for the problem is intimately associated with other important features of the plant. To gain a clear conception of this matter, we will divide the plant into the following elements:

(1) A great body of moving water, either completely confined in pipes or partially confined in more or less open channels.

(2) A turbine wheel so set that all the water passing through the pipes or channels must pass through the wheel.

(3) A gate, usually set immediately before the wheel, for controlling the power output of the wheel.

(4) A governor for automatically operating this gate so as to maintain uniform speed on the wheel shaft.

5. The ultimate function of the gate is to control the power output of the wheel. It does so (or attempts to) by varying the aperture through which the water flows. It may appear at first sight that this means would directly accomplish the purpose, but a little consideration will show that it will not do so in the manner required under modern conditions.

6. The gate consists, essentially, of an adjustable aperture, as stated above. It is set so that the water passes immediately from the aperture into the wheel, this remark applying alike to cylinder, register, wicket and other forms of gates as found on water wheels of the present day. Reducing the aperture (closing the gate) is intended to reduce the power output of the wheel, and increasing the aperture (opening the gate) is intended to increase the power output.

7. It will, however, be obvious that simply *increasing* or *decreasing* the aperture will not cause the power developed by the wheel to vary in like manner for the reason that any sudden restriction in the aperture cannot instantly check the velocity of the mass of moving water extending throughout the whole hydraulic system. The immediate effect of reducing the aperture is to cause the *same mass* of water to be ejected on the wheel, but with a *higher velocity,* thus actually increasing the power of the wheel at the very moment when uniformity of speed demanded that it should be decreased. On suddenly increasing the aperture, the reverse effect takes place.

8. It makes little difference if open channels are substituted for a closed pipe (except that certain danger elements are absent and an open channel usually has a larger cross section resulting beneficially in lower velocity of water), for the water will rise

or fall at the forward end under similar circumstances and produce similar results at the wheel.

9. Close regulation requires an exact balance, at every instant, between the power output and the load. Any lack of balance in this respect will produce an immediate effect in the speed, unless the energy of the revolving mass is considerable, compared to the energy in output per second. As a 1,000 horse-power wheel develops 550,000 foot pounds of energy per second, if the gate is unable to control the power output closer than within, say, 1 second of the instant that the gate is adjusted, the wheel must inevitably be accelerated or retarded an amount equivalent to this amount of energy. What the actual degree of this variation will be, will depend upon the momentum (or, more strictly, kinetic energy) of the mass to which the energy is applied, according to well-known laws. Unless this mass is very large (much larger than is usually found in practice) the speed variation from this cause alone will exceed the limits within which it is desirable to keep.

10. In plants where the pipe-line is long and the velocity of the water high, the gates cannot control the power output of the wheel within several seconds of the instant that the gate is moved, and when the energy of the revolving parts is low (as it frequently is, consisting of nothing but the momentum of a light water wheel and comparatively light revolving field or armature), no amount of perfection in a governor can prevent considerable momentary variations in speed when large changes of load occur suddenly. For it is to be observed that the causes of this variation lie quite outside the governor itself, as the variation would occur if the governor was instantaneous in its operation, and absolutely synchronous with the load change, conditions which can only be approximated in practice.

11. It was stated above that the act of varying the aperture through which the water passed not only failed to immediately control the volume, but actually produced a velocity in the water the reverse to that required, resulting in increasing the power output when a decrease was aimed at, and *vice versa*. This comes about because the gate is placed immediately before the water wheel, which takes up the energy of the stream. Attempts have been made to obviate this by placing the regulating gate elsewhere than immediately in front of the water wheel, though such arrangements have never come into general use.

12. One such attempt placed the regulating gate in the draft

tube, a valve of the butterfly form being used. This is quite an old idea, though some recent wheels have been equipped in this manner in one of the Niagara Falls plants. Althought it minimizes the deleterious effects referred to above, yet this arrangement brings with it its own troubles, the principal disadvantage being that the relation between angular movements of the butterfly valve and the power output of the wheel follows a very erratic curve, making the regulation of such a wheel by an automatic governor an extremely difficult matter. The wheels fitted with this system at Niagara Falls, referred to above, were provided with governors; but they were not operating when the author viewed the plant. Another attempt tried at Shawenegan Falls utilized an ordinary gate valve in the penstock, this gate being connected to an especially powerful governor. I have no information as to the success of this arrangement, but doubt its feasibility.

13. When the pipe-lines are long the use of relief valves or stand pipes mitigates the difficulty in some degree, as well as removes the danger to the pipe-line which would occur by a too sudden closing of the gate. A relief valve only takes care of one-half of the difficulty, for while it will check in a large measure the increased pressure due to closing the gate, yet it is powerless to avoid the decreased pressure due to an opening movement of the gate. A stand pipe is partially effective in both directions, if large enough, but only in a limited degree. When stand pipes and relief valves are provided, it is usually more on the score of safety than for regulation requirements.

14. The best forms of relief valves are undoubtedly those which do not depend in a rise of pressure to set them in action, but which are mechanically opened synchronously with the closing movement of the gate, afterwards gradually closing automatically. By properly proportioning the discharge of such a valve to the discharge of the water wheel, the gate can be closed very quickly without producing an appreciable rise in pressure. A plant at St. Catharines, Canada, equipped with such relief valves (made by Voith of Germany) and having pipe-lines several hundred feet long, head being 298 feet, has been found to give remarkably good results when suddenly throwing off large amounts of power. The exact figures are not obtainable, but I am told on good authority that the maximum momentary rise in speed on throwing off 75 per cent. of the full load was under 4 per cent., an extremely good figure for such a plant. The units are of 7,000

horse-power, revolving at a high speed and having a large momentum factor.

15. From what has been said it will be apparent that one of the obstacles to further improvement in speed regulation is inherent in the accepted designs of gates which are unable to entirely discharge the functions for which they are intended. Commercial reasons, however, and the existing organizations of manufactures, render it unlikely that any great departure in principle of gate construction will be made in the near future, though there is no question but that considerable improvement is desirable, not only for the reasons given above, but because water-wheel gates, as now designed, are poorly working apparatus at best, however well constructed.

16. We now have to pass to the consideration of the governor itself. The limits of this paper preclude more than a brief reference to its essential features and an examination of certain of the characteristics of its operation.

17. The most successful governors are of the hydraulic class. They consist of a centrifugal governor belted from the water wheel and adapted to move a small piston valve. This valve hydraulically controls a larger valve (which is too heavy to be moved by the centrifugal governor direct) and this in turn controls a piston in a hydraulic cylinder, the piston being connected by suitable mechanical means to the gate of the water wheel. The parts are so related that, if the balls collapse below their mid-position, the piston will open the gate, and if they extend beyond their mid-position, the piston will close the gate.

18. The governor contains also another important element (the compensator), the functions of which will be realized when it is borne in mind that the speed of the wheel is not restored, after its initial deflection, until an appreciable interval after the gate has been moved. This interval would be sufficient for the governor to move the gate to its full open, or closed, position and violent racing would result. The compensator checks this by causing the governor piston to move an amount proportional to the degree of the initial speed variation. This is accomplished by making the movement of the governor piston react on the centrifugal governor, or its connections, so as to virtually reverse the movement which was initially produced by the changed speed, induced by the changed load.

19. In America the manufacture of water wheels and of gov-

ernors has been carried on (in the great majority of cases) as separate businesses, and on account of the great diversity of designs of water wheels and their gate rigging, the governor manufacturer has perforce been obliged to adopt a design which is applicable to all designs of water wheels. This, for commercial reasons, necessitates the governor being self-contained and independent of the wheel, and usually results in an unavoidable complication in the connections between the hydraulic piston and the gate.

20. From an engineering standpoint, the European practice is to be preferred where the wheel and governor are incorporated in one design. The hydraulic piston is directly connected to the gate, thus avoiding the rotary shafts and geared connections necessary with the separate governor. In this country, however, with its tendency to specialize, the greatest success, both commercially and technically, has been secured when the water wheels and the governors are made by separate manufacturers who are specialists in this line. Some of the water-wheel manufacturers have attempted building their own governors, but they have usually abandoned their efforts for the reason given above.

21. In order to secure good regulation it is necessary, first, that the smallest departure of the speed from normal shall cause the hydraulic piston to operate, and, secondly, that it shall operate with extreme quickness. Even if after full load has been thrown off the governor occupies one second in closing the gates, a considerable excess of energy will be produced (amounting to over half a million foot pounds in a 1,000 horse-power wheel) which will cause a considerable acceleration unless the momentum of the revolving elements is very large. In many cases it is not even practical to close a gate as quickly as this for mechanical reasons; and having arrived at a practical limit in this respect, the only possible way of preventing considerable momentary fluctuations when large changes of load occur, is by making the momentum of the revolving element sufficiently great to take care of the excess energy. This point is very rarely considered by the designers of water-power plants.

22. During some tests conducted by the author on a 1,100 horse-power unit, on throwing off the full load, the five gates were completely closed in $1\frac{3}{5}$ seconds, an extremely quick performance when it is borne in mind that 96 leaves in the five gates had to be moved a considerable distance (through moving water) in addition to the

mechanism for operating them, the moving parts of the governor and the heavy connections between governor and gate shaft, as well as a counterweight weighing nearly one ton. Even under these

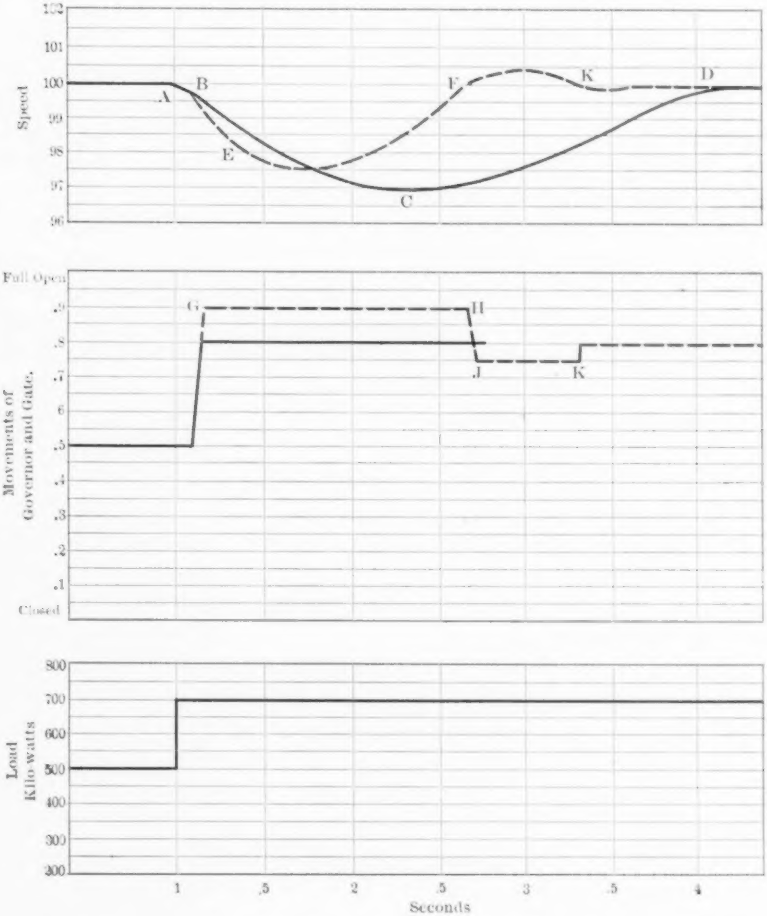

FIG. 1.—SHOWING DIAGRAMMATICALLY, IDEAL RELATIONS BETWEEN SPEED, GOVERNOR MOVEMENTS, AND LOAD CHANGE UNDER EXISTING CONDITIONS.

circumstances the wheels and the revolving field of the alternator (weighing several tons) accelerated over 7 per cent. in this short interval, the speed rising from 120 to 129. This was partly due to the energy developed by the wheel during $1\frac{2}{3}$ seconds while the

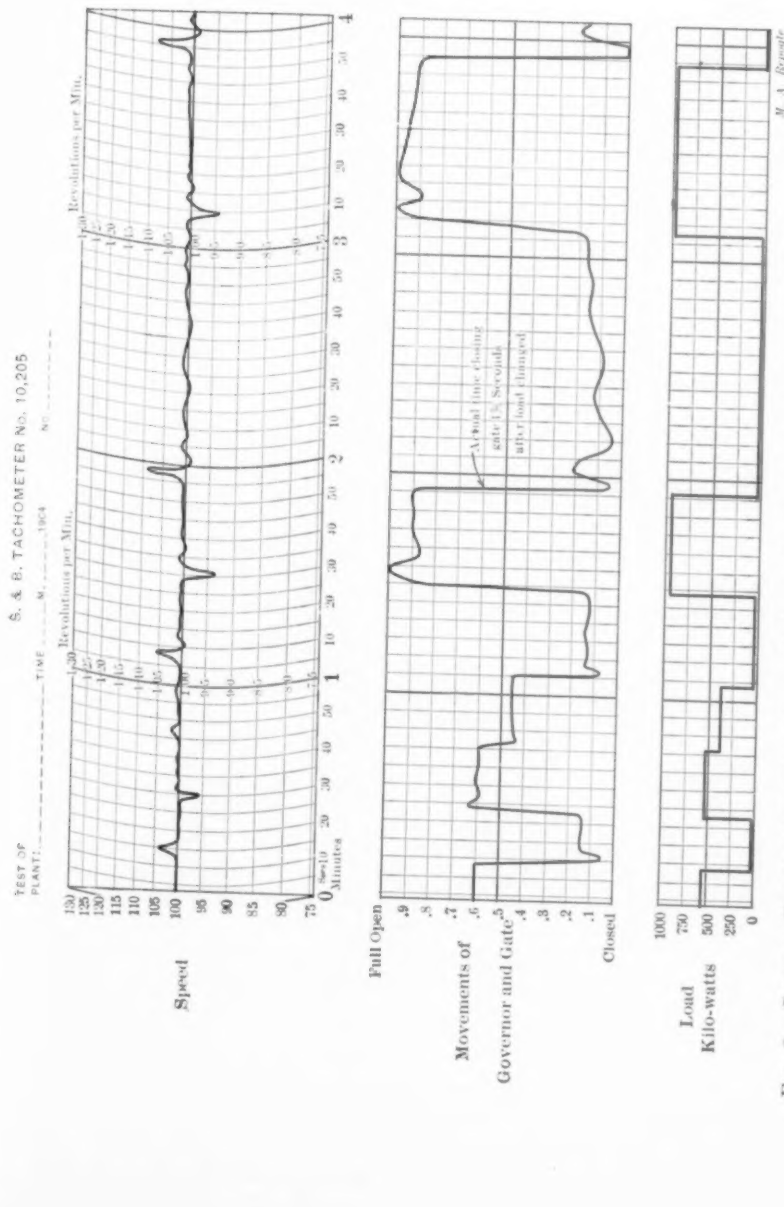

FIG. 2.—CURVES, REPRODUCED FROM TESTS, SHOWING RELATION BETWEEN SPEED, GOVERNOR MOVEMENTS, AND LOAD CHANGE, WITH HEAVY SUDDEN CHANGE OF LOAD.

gates were closing, and partly to the inertia effects of the water referred to in the former part of this paper.

23. On watching the forebay while the experiment was repeated, the water was seen to rise several feet, and a great turmoil took place, though the channels were capacious and short. The effect

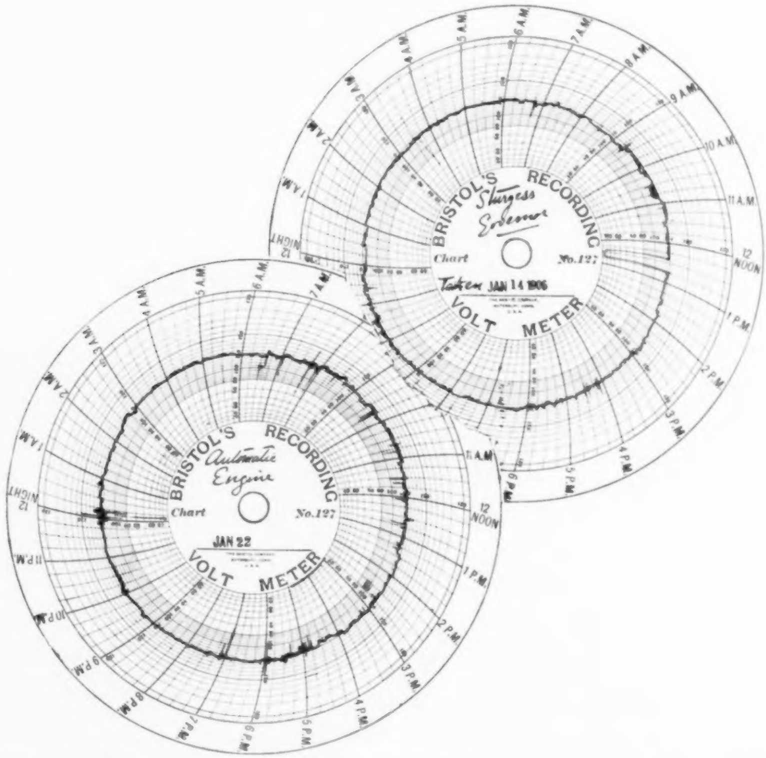

FIG. 3.—VOLTAGE RECORDS, SHOWING COMPARISON BETWEEN REGULATION SECURED BY STEAM ENGINE AND WATER WHEEL GOVERNOR.

on the wheel was, of course, violent fluctuations in effective head. The speed was maintained fairly uniform, though the governor necessarily continued moving the gate as the head fluctuated, producing, at first sight, the appearance of racing.

24. A similar test conducted at another plant where the head race was several hundred feet long, and partially closed at both ends by the head gates and intake, resulted in the formation of a long low wave which continued to travel back and forth in the

channel for a great length of time, creating at the wheel a rythmic rise and fall of head of about 20 per cent., having a period of several minutes. The governor responded by slowly opening and closing the gates as the head fell and rose, and at first sight this was thought to be racing (a slow form peculiar to water-wheel regulation) as the load was uniform.

25. The diagram, Fig. 1, indicates, in general character, the greatest perfection that can be obtained with any governor under prevailing conditions. The upper curve indicates the speed, the

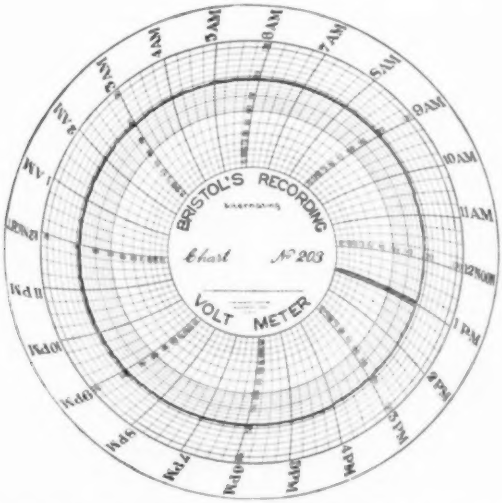

FIG. 3A.—VOLTAGE RECORD OBTAINED FROM WATER WHEEL—LIGHTING LOAD

middle curve shows the movements of the governor and gate, and the lower curve the load.

26. Following these curves simultaneously we note that at the start the speed is normal, the gate half open and the load 500 kilowatt. At the moment indicated by 1 second the load is suddenly increased to 700 kilowatt. As a first effect the speed immediately begins to fall, following the line *A-B*. After the fall has reached, say, 0.1 per cent. the governor is thrown into action, and the gate is suddenly opened to 0.8 gate.

27. For reasons that have been already outlined, this opening movement of the gate will not cause the wheel to instantly inrease its power. Therefore, the speed will continue to drop, following the curve *B-C*. As the water column accelerates, the power of

the wheel will increase and the speed rise, following the curve *C-D*, being finally restored to normal (providing the gate was moved the right amount).

28. It is to be noted that, with the exception of the slight drop in speed, A to B, which took place before the governor moved the gate (say, 0.1 per cent.) and the consequent delay in moving the gate after the load changed, the subsequent drop in speed is due to causes quite outside the governor itself.

29. In some cases, however, by adopting an expedient, an im-

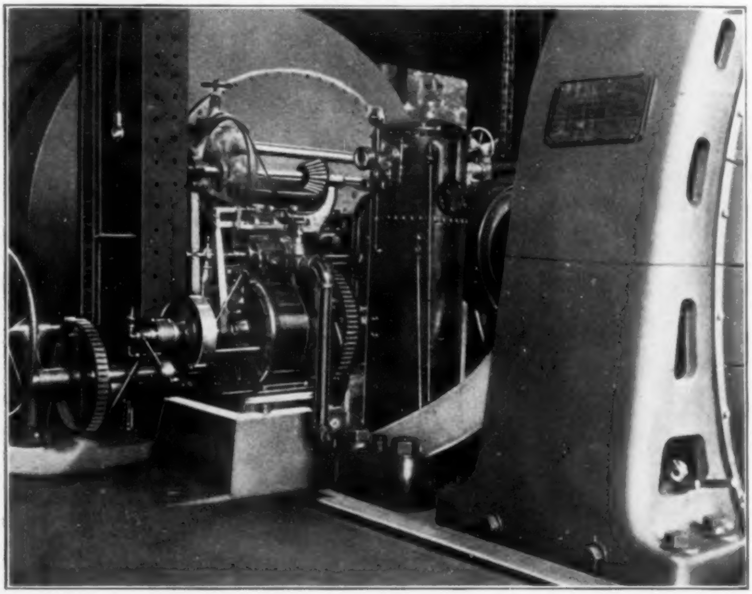

FIG. 4.—30,000 H. P. PLANT OF THE HUDSON RIVER ELECTRIC COMPANY. SPIERS FALLS. 5000 H. P. UNITS. 20 FT.-TON GOVERNORS.

provement can be effected. Thus, if instead of stopping at 0.8 gate (the amount actually necessary) the governor had continued opening the gate to 0.9, as shown by the dotted lines *G-H* on the diagram, the acceleration of the water column would have been augmented, the speed following the curve *B-E-F*. The gate being too far open, however, the speed will rise above normal, but at this moment the governor will make a quick reverse movement *H-J*, turning the speed curve downwards (*F-K*). A third similar move-

ment may take place later. The dotted curve is an obvious improvement over the full-line curve.

30. The curves (Fig. 2) show some tests made by the author on a 1,100 kilowatt unit. The curves were traced by a special Schaffer and Budenberg Tachometer, the readings being sufficiently magnified to bring out the characteristics of the governor. The reverse curves, similar to the dotted curve on Fig. 1, are clearly seen. The load changes and governor movements are plotted below. Note that when the whole load was thrown off (at 1 minute,

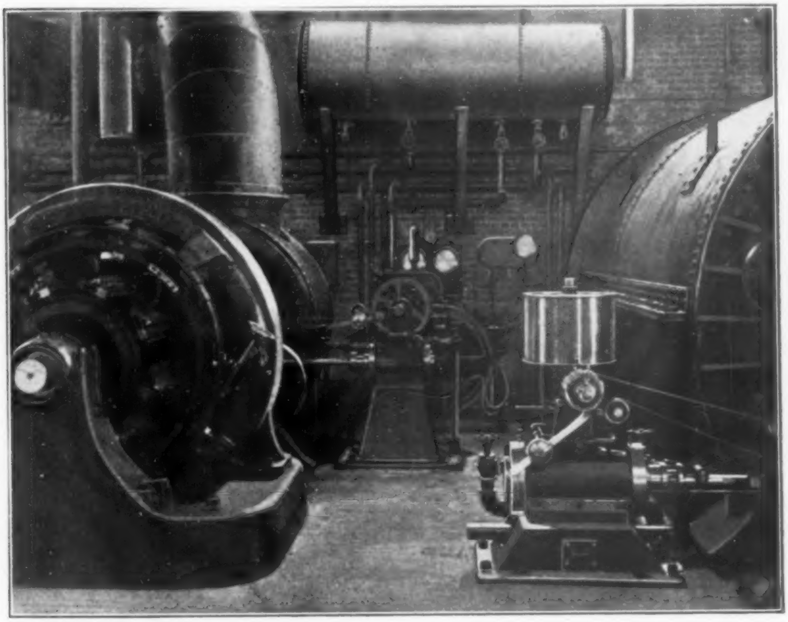

FIG. 5.—12,000 H. P. PLANT OF THE HURONIAN COMPANY, CANADA. THE HYDRAULIC GOVERNORS ARE OPERATED BY A CENTRAL PRESSURE SYSTEM SUPPLIED BY DUPLICATE MOTOR-DRIVEN OIL-PUMPS.

55 seconds) the speed accelerated about 8 per cent. in an incredibly short time (under 1 second).

31. The governor had the gate shut in 1.4 seconds after the load went off, and its subsequent behavior, as shown by the middle curve, was interesting in view of what has been said before. It is to be noted that after the first quick reverse at 2 minutes the governor slowly oscillated for about another minute, but with a gradually increasing gate opening, the speed and load being prac-

tically constant. This was due to the water rising in the forebay and gradually subsiding in a succession of waves, the governor taking care of these fluctuations in effective head in a very intelligent manner. A similar effect is shown at 3 minutes 10 seconds when the load was thrown on.

32. A comparison of the regulation (as gathered from a voltage record) of a water wheel and a steam engine is shown by the Bristol records, Fig. 3. The engine is of the Ball and Wood type, having an automatic cut-off controlled by a good inertia-centrifugal governor. For all practical purposes the records are equal. The load was principally induction motors of various sizes. These curves are not offered as the best examples of speed regulation that can be obtained, but as showing the characteristics of water-wheel governing. It is quite possible to get charts which simply show a straight uniform speed line such as Fig. 3 A, but nothing is to be learned from such, unless the load conditions are known.

33. When attempting to forecast the degree of regulation obtainable in a given plant, we find that one of the most important items, that is, the maximum momentary variation which will occur when making given changes in load, is practically uncalculable. An equation can be devised for ascertaining this, but solving it calls for data which have not yet been procurable. Under the circumstances all we can do is to compare the plant with others of similar design, on which accurate tests have been made, and make our forecast accordingly.

34. We can, however, forecast with precision the following, and if this is realized, and the plant properly laid out, very good regulation will be obtained under ordinary conditions of operation:

(1) The governor shall apply to the gates a definite amount of energy.

(2) It shall operate on the speed varying ½ per cent. from normal.

(3) On a considerable change of load taking place it shall promptly move the gates to the position required by the changed load.

(4) It shall move the gates through their full range in 1½ seconds.

(5) Under steady head and steady load it shall remain stationary, and it shall not make more than three movements in readjusting the gate after any change of load.

35. The necessity for the first clause, instead of a definite

statement that it shall move the gate, arises from the fact that the amount of energy required to move a gate is a very uncertain quantity, for which no one seems to be held responsible. A 5,000 horse-power wheel may require from 15 to 20 foot tons to move the gate. One such wheel with a flanged gate operating under 78 feet head required 34 foot tons when it was in good condition and 34 + x when it was in bad condition, x being entirely indeterminate, as it was caused by the water wheel resting in the

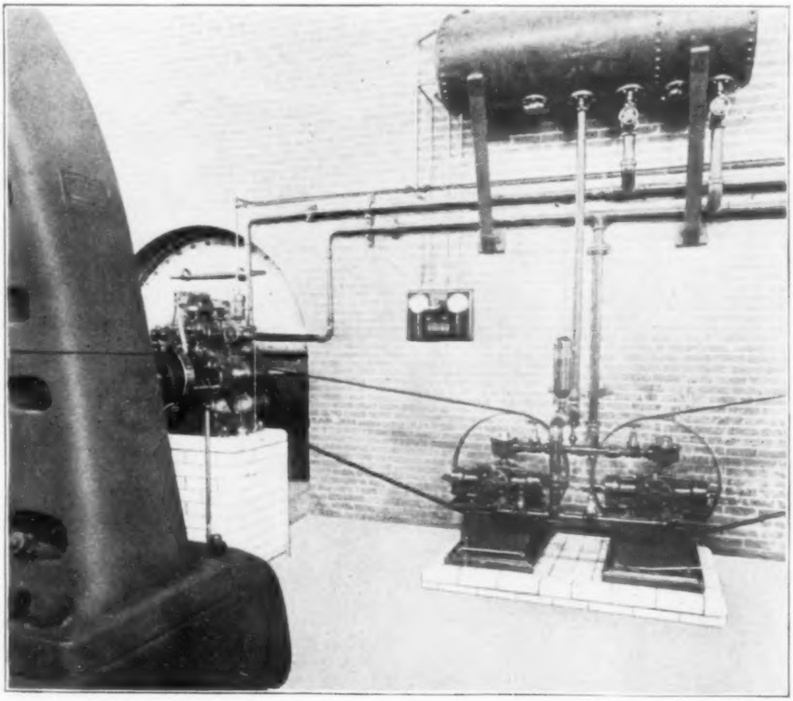

FIG. 6.

gate and raising ridges in the metal, which effectually prevented the gate being moved one way or the other.

36. The general arrangement of the governors on the 5,000 horse-power wheels of the Hudson River Electric Company's plant at Spiers Falls, N. Y., is shown on Fig. 4. The governor shaft is connected to the gate mechanism of the water wheel by a cable drive. The storage tank, holding the oil under pressure, is seen

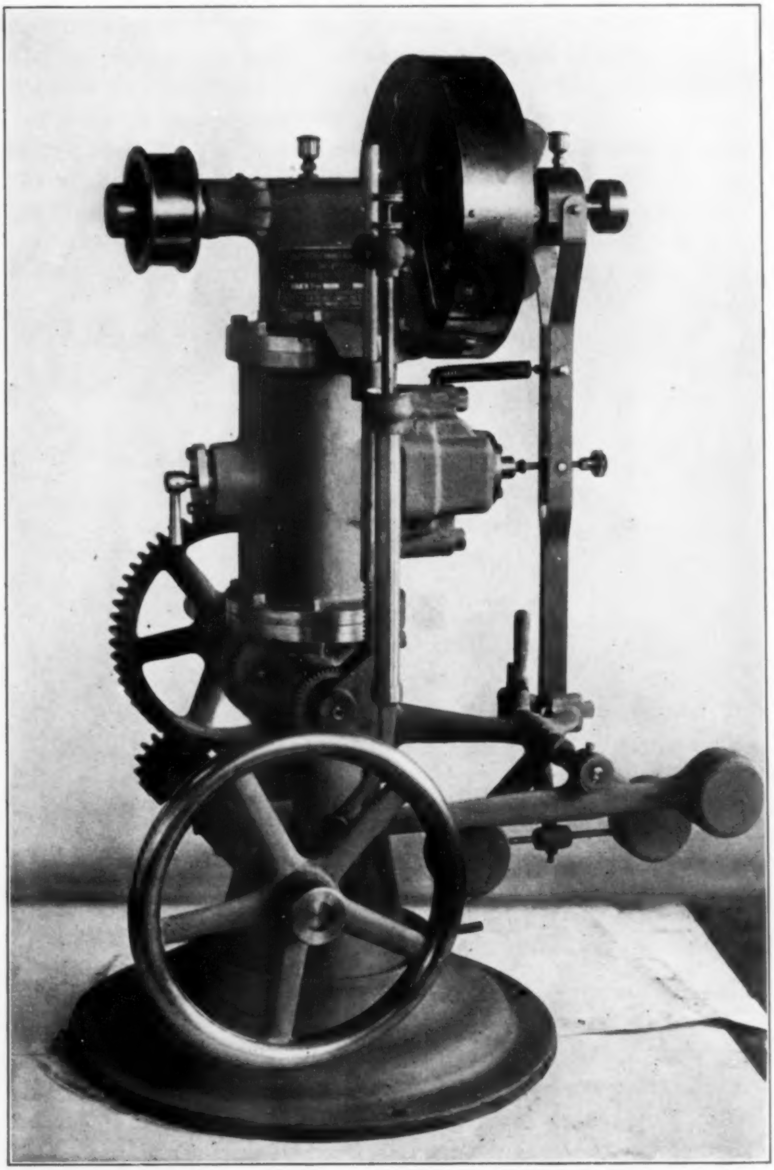

FIG. 7.—VERTICAL FORM OF WATER-WHEEL GOVERNOR.

immediately behind the governor, the pump which maintains the pressure being placed on the opposite side of the wheel shaft. Other arrangements are shown on Figs. 5 and 6, where one large tank was used to supply the governors, either a number of belt-driven pumps, or one motor-driven pump being used to maintain the oil in the pressure tank. This arrangement has been used by the author in many instances.

37. A later design of governors is shown on Fig. 7. The reciprocating piston rotates the governor shaft by means of a rack and pinion concealed within the pedestal of the governor, forming practically an extension of the cylinder. An important feature in this machine, not shown on view, is the arrangement of conical-lift valves to give admission to the cylinder. These replace the piston valve which has hitherto been used exclusively. However, time will not permit us to discuss these features or others belonging to the constructional details of the governors generally.

38. In conclusion, it may be said that the regulation of water-power plants, one or two aspects of which we have considered above, is one of supreme interest and importance, and is in need of more accurate data than the author has been able to obtain in a somewhat active life, devoted largely to the business end of the problem.

No. 1113.*

TURBINE DESIGN AS MODIFIED FOR CLOSE REGULATION.

BY GEO. A. BUVINGER, DAYTON, O.
(Member of the Society.)

1. The American Manufacturers of water wheels have, as a rule, adopted the practice of connecting to their turbine a governor which is guaranteed by its builder to meet the requirements of the electrical machinery with which it is to be installed. For it is the demands of the electrical plants, with their heavy changes of load at constant speed, that have led to the high development of the modern governors. This practice, which follows the spirit of specialization now so common, has left to the turbine builders the problem of fitting their wheels with gates that are as nearly balanced as possible, and are connected to the governor by simple and strong gate work. As a result of their experience, they have labored to keep all gears and wearing parts on the outside of the flume, where they are removed from the grit in the water, and are easily accessible for observation and repair.

2. There are many types of gates now in use, some of which are so nearly balanced that they can be used under great variations of head, while others are only suitable for low heads. As each type has its advocates among the engineers, who recommend and approve its use, the majority of manufacturers have been led to adapt their wheels to meet the specifications submitted, although remaining partial to the type most generally supplied with their standard wheels.

3. Two types that have had the most universal use with the inward flow reaction turbine are the cylinder gate and the swivel or wicket gate.

4. The cylinder gate in its simplest form is a plain iron ring,

* Presented at the Chattanooga meeting (May, 1906) of the American Society of Mechanical Engineers, and forming part of Volume 27 of the *Transactions*.

fitting closely on the inside of the chute case, as shown in Fig. 1, and opening a part of all the chutes as it moves parallel to the axis of the shaft. Its part gate efficiency is not as good as that of the swivel gate, but it is subject to little wear, remains tight after long usage and is practically balanced when on a horizontal wheel. When the wheels are set horizontally, it is very important that a support should be provided for the gate, as

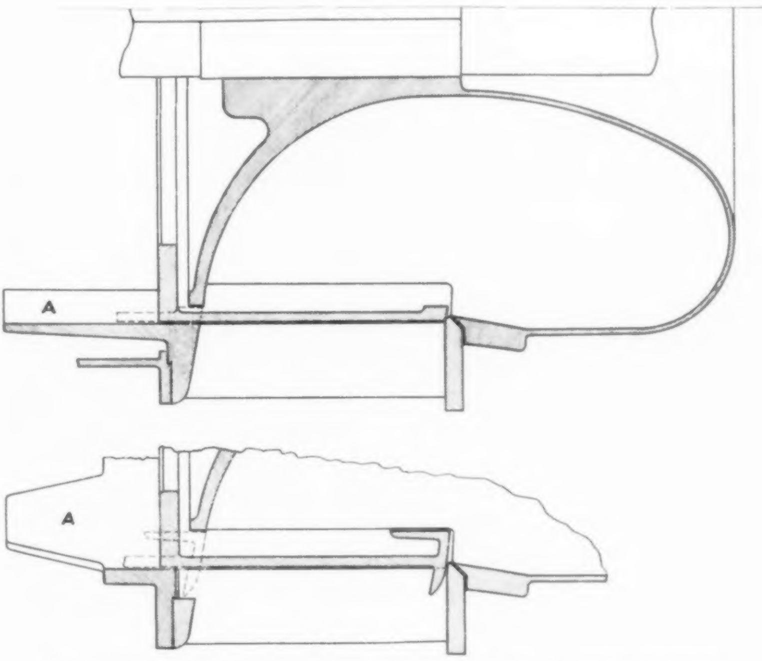

FIG. 1.—UPPER SECTION—PLAIN CYLINDER GATE. LOWER SECTION—CYLINDER GATE WITH LIP.

shown at "A," or the gate will tip as its center of gravity moves beyond the case, and there will be a decided tendency of the gate to bind in opening and closing beyond this point. This can be accomplished by casting guides on the case, as shown in the cut, and finished when the case is being tried. This gives the best form, but guides are sometimes fitted on the dome. In some cases when the best results have been desired these guides have been lubricated by forcing grease under the gates by means of grease cups. For vertical wheels its weight must be counter-

balanced, which is easily accomplished, but care should be taken that the counterbalance weights have a slow velocity in order that their inertia does not have a detrimental effect on the governor. For quite a number of years the practice has been followed of adding lips to these gates, as tests have shown that it improves the part gate efficiency to some extent. It has only been lately, however, that the governor manufacturers have cared to govern these gates under the higher heads now developed, as they are much harder to open, having a great tendency to close of themselves. Al-

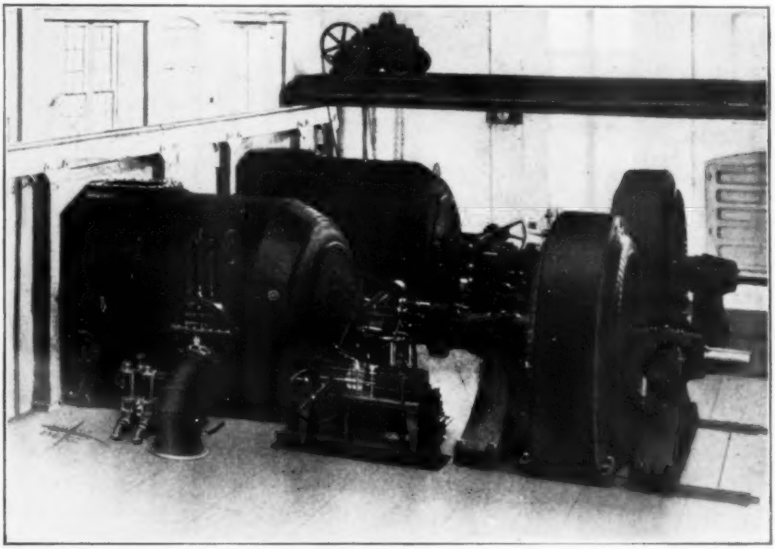

Fig. 2.—Two pairs of Cylinder Gate Wheels, with Governor and Relief Valves, developing 4000 H.P. each under 68 feet head.

though they have often been used it is no unusual thing for them to be chipped off when in place to relieve the governor of more work than it could do. At the present time there is being installed a plant having four units of four pairs of wheels under 50 feet head having lips on the gates, and the company furnishing the governors have guaranteed the usual regulation. These gates when moved by means of rods in the wheel case and rack and pinion outside have given great satisfaction. In fact, one of the governor companies has stated that there is no instance in its long experience where it has failed to obtain the highest degree of

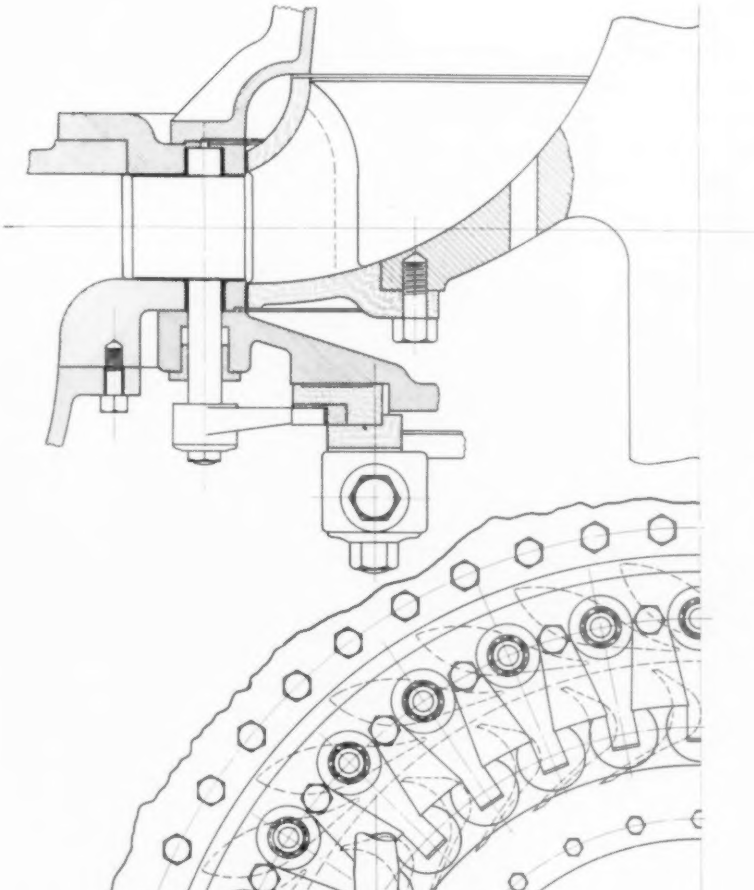

FIG. 3.—SWIVEL GATE.

speed regulation in connection with the plain cylinder gate of the type shown. Fig. 2 shows two pairs of cylinder gate wheels developing 4,000 horse-power under 68 feet head.

5. The swivel gate when properly proportioned can be almost

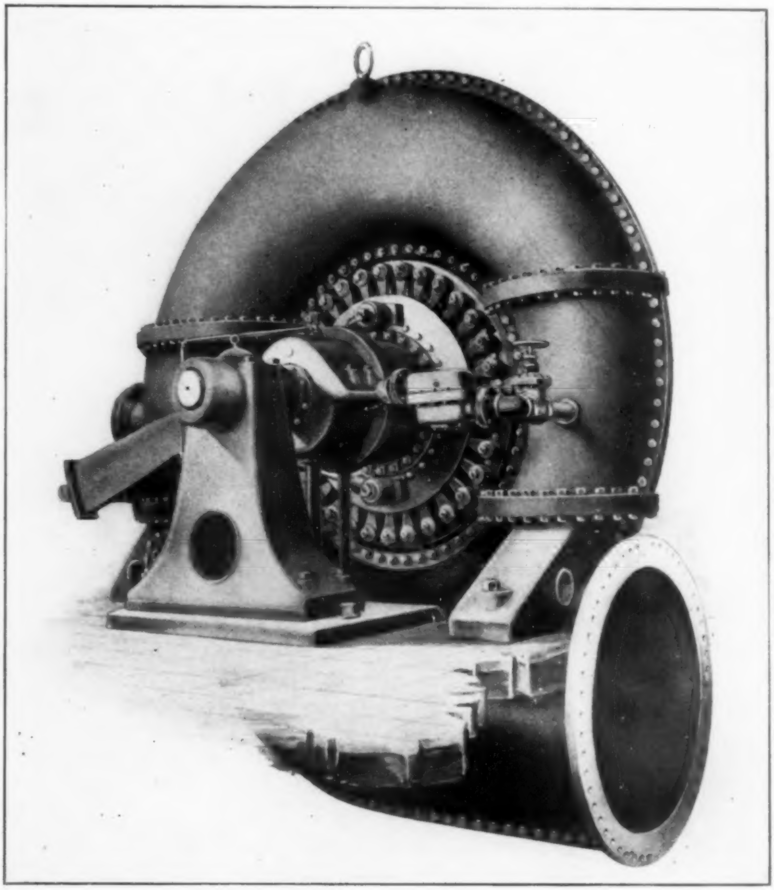

FIG. 4.—SINGLE SWIVEL GATE WHEEL, SHOWING GATE WORK, DEVELOPING 9000 H.P. UNDER 270 FEET HEAD.

entirely balanced, and a slight tendency to close may be considered a benefit should any part of the governor or connections become damaged. The part gate efficiency of this type of gate is very good, as the opening of each chute is reduced in such a manner that there are no eddies formed as in the cylinder gates. For

high-head work the type shown in Fig. 3 has given good results. The gate and trunnions are cast in one piece of cast steel, and the trunnion extends outside of the wheel case, where it is connected by means of levers to the moving ring and gate work. It is planed to template, insuring accurate fit, and tight closing, and there is no obstruction to a smooth flow of the water. These two types of gates have been used under all heads from 5 to

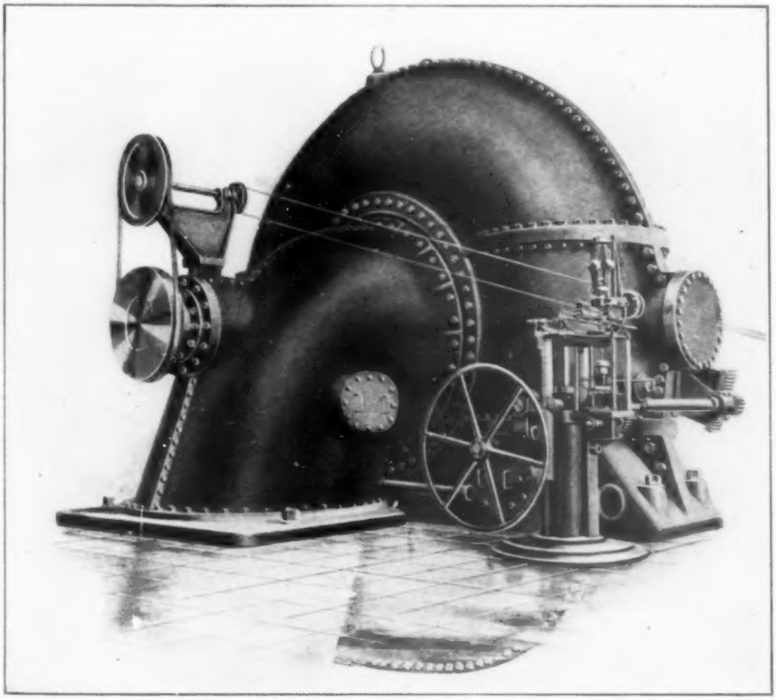

FIG. 5.—SINGLE SWIVEL GATE WHEEL, SHOWING GOVERNOR, DEVELOPING 9000 H.P. UNDER 270 FEET HEAD.

500 feet and have stood the hardest kind of wear. Figs. 4 and 5 show a swivel gate which, in a spiral flume, develops 9,000 horse-power under 270 feet head.

6. The development of powers under heads from 100 to 1,000 feet where the quantity of water is small has led to the use of the Girard type of impulse wheel. This did not adapt itself to the use of either of the above types of gates, and that shown in Fig. 6 was adopted.

7. This gate by uncovering one chute at a time gives a very good efficiency curve, as only one, two or three chutes are subjected to the losses caused by disturbance of the stream. These

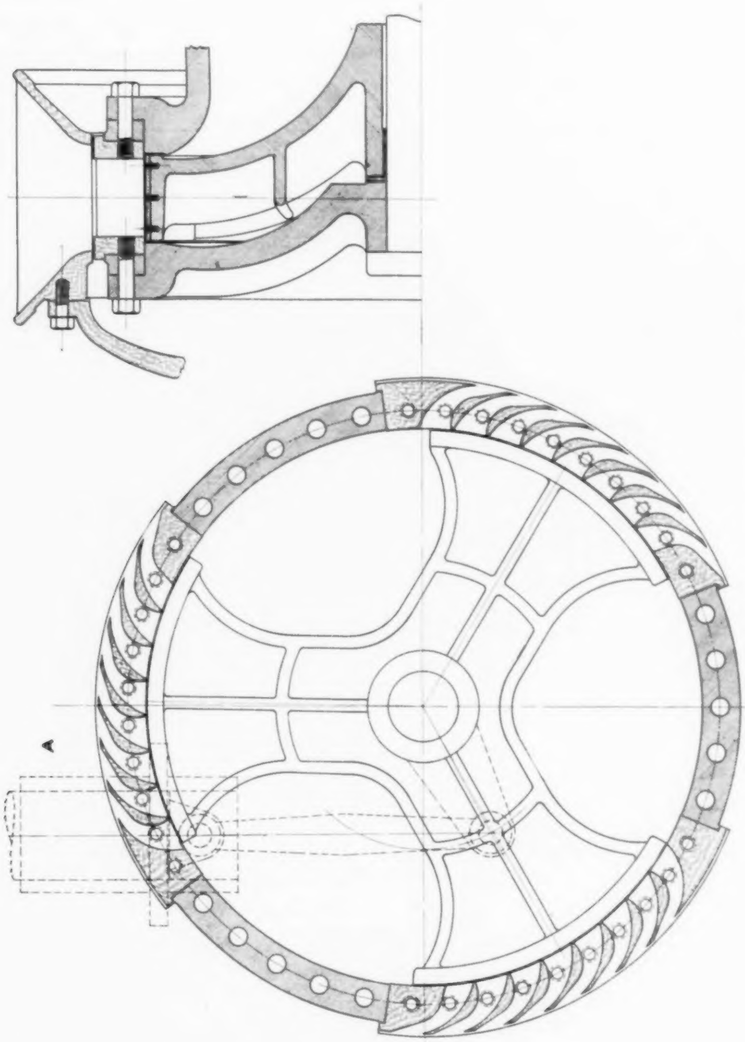

Fig. 6.—Balance Gate for Girard Wheel.

gates under high heads showed a great tendency to close, and an equalizing piston, shown at "A," was introduced, by means of which this can be reduced to any amount and the gate balanced.

Fig. 7 shows a Girard wheel developing 2,000 horse-power under 630 feet head.

8. The connections between the governor and gates, no matter what type is used, require very careful consideration in their design. All gears should have cut teeth, and the shafting should be of sufficient strength to resist twisting, for it is very important that there should be no lost motion between the governor and gates. The many and sudden changes of load with a quick-acting governor throw the worst possible strains on all the parts, and

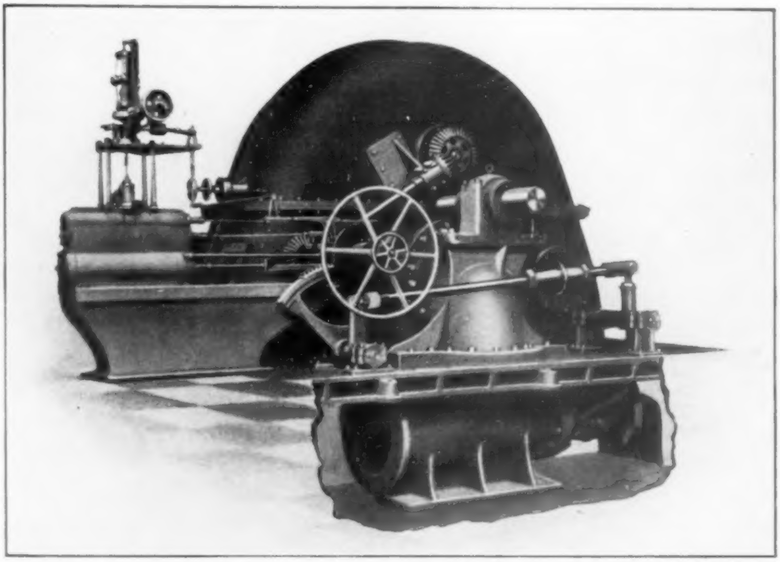

FIG. 7.—GIRARD WHEEL, SHOWING BY-PASS AND GOVERNOR, DEVELOPING 2000 H.P. UNDER 630 FEET HEAD.

they must be strong enough to stand this. This strength must be accomplished, however, without increasing the mass of the moving parts any more than absolutely necessary, as the inertia of these rapidly moving parts is very great. If these points are correctly designed the governor will be able to keep the variation within 2½ per cent. under changes amounting to half the load when the power-house conditions are favorable.

9. The points now remaining to be considered are the draft tubes and feeders, and these are often beyond the control of the builder, depending on the layout determined upon by the en-

gineer, who must be more or less governed by local conditions. The regulation is often rendered difficult by the character of the draft tube, which, if incorrectly designed, will cause a great deal of trouble. The vacuum or draft head attained in the draft tube is a variable quantity, depending on the quantity of water discharged by the wheel. The area of the draft tube should be such that at full gate the draft head is approximately equal to the distance of the shaft above tail water less the resistance in the pipe and the velocity head of the water leaving the draft tube. To decrease this velocity as much as possible the diameter of the draft tube is often increased at the lower end. As the quantity discharged decreases from full gate to part gate the vacuum head also decreases, and if the change is very rapid the vacuum may be entirely broken, especially if the draft head is large. To decrease this tendency as much as possible the draft tube should be so designed that the water at the top has the same velocity as the water leaving the runner and the tube then tapered to the bottom, so that the water will leave with a velocity of from two to three feet.

10. When wheels are placed in an open flume, or with a very short feeder from the turbine to head water, and a correct draft tube, everything is favorable for good governing.

11. In a large number of plants, however, the conditions are not so simple, and it is necessary to overcome certain objectionable features in order to get the best regulation. This is the case when the water is supplied to the turbines through a long steel feeder, for the inertia of the long column of water must be overcome in order to supply the changing quantity at the gates, and the feeder itself must be protected from the violent shocks.

12. The action of the water under the above condition is about as follows: When the unit is running at normal speed a sudden drop in load will cause the speed to increase and the governor will start to close the gates. The compensating device will retard this action somewhat to allow the velocity in the feeder pipe to change, but when the pipe is quite long this will not be sufficient and there will be a sharp rise in pressure or head which will make the gate opening still too large for the load, and the governor will close still farther until the head drops to normal, when the opening will be too small and the gates will open. This causes an alternate rise and fall in the speed and the governor will oscillate until it finally comes to its correct position. When a load is thrown

on the generator the speed will decrease, causing the governor to open the gates; but as the velocity of the long column of water cannot be instantly increased the head will be decreased and the gates open farther than necessary under normal head. As the velocity is gradually accelerated in the feeder the head will rise and the power and speed increase, causing the gates to close. There will then be a similar oscillation to that occurring when the load falls before the governor has reached its correct position. As very large and frequent changes of load take place in some plants, the governor may be almost constantly on the move, causing variations in speed and voltage that naturally affect the efficiency and value of the plant. An even more dangerous result of this action is the variation of pressures in the feeder itself, giving rise to breathing of the pipe and, in cases where there are a large number of bends on the line, causing the pipe to writhe and twist. This will strain the joints, causing leaks, and may result, in some severe cases, in a disastrous break. Although the remedy for this is usually applied direct to the feeder, there are some appliances that may be added to the turbine, and consequently the consideration of the best method is a point that affects the turbine builder. When the conditions are not too severe it is sometimes possible to overcome these defects by means of flywheels and air chambers.

13. The flywheel, by giving a portion of its energy to the turbine, will extend the length of time during which the governor will open or close the gates and thus reduce the severity of the changes. This is not sufficient, however, for most cases and is seldom used, especially as the generator has considerably more flywheel effect than would be added by most engineers. Another partial remedy is the addition of an air chamber on the turbine casing in order to relieve the changes in pressure. When we consider the fact that the capacity of any air chamber which it would be practical to add is so slight in comparison with the capacity of the feeder we can see that this is not sufficient for any very severe strain. Moreover, the water will exhaust the air from this chamber and it will have to be restored by an air pump and may be useless when most needed.

14. The most successful remedies are those that are applied strictly to the feeder pipe, and although they do not come directly under the notice of the builder in many cases, still, when he also supplies the feeder pipe, they are a part of his contract.

Moreover, it is his duty to protect his wheel from severe shocks and to make the conditions as favorable as possible for the governor which he is supplying.

15. The addition of a stand pipe to the feeder has given better results than any other method and has been used successfully under heads as high as 250 feet. It should be placed as near the power house as practical and should be of sufficient size to supply water to the plant during the time necessary for the column of water in the feeder pipe to change from one velocity to the other under changes of load as high as 50 per cent. at least. This water should be supplied without drawing down the head in the pipe more than a few feet, and the pipe should be supplied with an overflow a foot or two above the level of the head water. When thus designed, the head will not vary greatly in the feeder pipe and we will obtain a rapid and close regulation with perfect safety to the feeder. In northern climates great care must be taken to keep the stand pipe open and free from ice in winter, and this is usually done by lagging and sometimes by the introduction of steam pipes into the stand pipe. In several plants installed with no protection to the feeder very grave defects have been overcome by adding this pipe after the plant had been in operation. If the feeder is almost vertical at the power house and then has a long portion with little slope, a much shorter pipe can be put at the beginning of the drop with very good results.

16. Where the head is so great that a stand pipe is not practical, it is sometimes possible to arange a by-pass that opens as the gates of the turbine close. There is no objection to this in places where the water is used for irrigation and must be supplied whether used by the wheel or not. With this device operated by the governor the velocity in the feeder remains constant, and regulation is very simple. It is not, however, economical where we wish to store the water, as in many plants where the water is scarce. The arrangement of a by-pass with governor control is shown in Fig. 7.

17. In such cases we can simply protect the pipe against the rise in pressure by adding relief valves which keep the pressure within a few feet of the normal head, but can do nothing to accelerate the water when the gates open. These operated by hydraulic pressure regulators are very satisfactory, but whether of this type or of the spring type they should be tested at least once a day, so that they are sure to be in working order when needed.

18. The experience at the West India Electric Company's plant

may give a good idea of the relative value of some of the points above discussed. The plant of the West India Electric Company, installed in 1898 near Spanish Town, Jamaica, probably presented as great difficulties as any erected. The installation consisted of a power house, two generator units of 400 horse-power, each under 56 feet total head, two exciter units, a single steel feeder, 6,200 feet long and 8 feet in diameter, a skimming basin, and an 8-foot concrete dam. The character of the country was such that the feeder pipe, besides having a large number of bends, had long sections with very light slope. The first 1,250 feet had a slope of only one foot, then a fall of 100 feet in 250 feet, followed by a section 1,350 feet long with one foot slope. At the end of this portion there was a rise of 10 feet in 250 feet, and the remainder of the pipe had a slope of 27 feet of varying degrees.

19. A stand pipe 8 feet in diameter and 85 feet high was placed at this lowest point, and beyond this there was a rise of 6 feet in 50 just outside of the power house. The 47 feet effective head was obtained with a draft head of 28 feet through long draft tubes. The two generators were each driven by a pair of 21-inch cylinder gate wheels at a speed of 400 revolutions per minute, and controlled by a Giesler Electric Generator. On operating the plant it was found that although the governors were able to keep the speed within the 5 per cent. variation guaranteed with change of 50 per cent. load, the stand pipe was entirely inadequate to relieve the feeder pipe from the strains set up by the varying loads. At points there was a decided flattening and breathing that caused a great strain on the joints. The capacity of the 8-feet stand pipe was 41½ cubic feet per foot of fall, while a 50 per cent. variation of load on the two generators caused a call for 56 cubic feet per second, so that the storage effect of this pipe was entirely too small. A plate steel flywheel, weighing 10,000 pounds, was added in order to extend the time of change of speed, but this was also insufficient. As soon as it was seen that the only remedy was the addition of storage capacity on the line, the feeder pipe was extended beyond the power house and carried to a steel tank 50 feet in diameter and with a depth of 12 feet below the level of the dam. The capacity of this tank, 2,000 cubic feet per foot of fall, or enough to supply the water for 35 seconds for a change of 50 per cent. load on both wheels, was more than sufficient to allow the column of water to overcome its inertia, when more water was

called for, by opening the gates. After this change was made the trouble was removed and the entire plant was satisfactory.

20. In the above, the writer has kept out any discussion on the merits of the different governors now on the market and has left the description of details and operation to the builders themselves. These governors are the outgrowth of long experience with the difficulties to be met in this line, and the excellent results obtained are the best testimonial of the care and study which have been given to their development.

21. These results assure us that, with a correct understanding between the engineer, the turbine builder and the governor manufacturers, the most severe conditions can be met in such a manner as to obtain the closest regulation demanded in any plant. To arrive at this result it only is necessary to determine the following factors:

1. The amount of power required to open and close the gates.
2. The time necessary to close the gates without shock or strain to any part of the plant.
3. The flywheel effect of the revolving parts of generator and turbine.
4. The effect of the draft tube.
5. The character of the load.

No. 1114.*

A NEW LIQUID MEASURING APPARATUS.†

BY GEORGE B. WILLCOX, SAGINAW, MICH.
(Junior Member of the Society.)

1. The liquid measuring apparatus herein described was originally designed to automatically measure brine flowing from a salt well into a salt works. After about a year's satisfactory operation in that capacity, tanks of the same type, but modified in detail, were designed and set up as a permanent part of a boiler-room equipment, to automatically measure and record the amount of feed water supplied to boilers. Apparatus of this kind designed as a portable testing device has proven of the greatest service on evaporative tests, for it does away with the uncertainties and hard work of weighing by the usual tank and scale method.

It may be more correctly termed a weighing rather than a metering device, as will appear later, and it will be so described.

2. Notable features of the device are: Elimination of constant errors of graduation, present in many water meters. Its operation is not appreciably affected by air in the liquid, or by changes in temperature, head or quantity of discharge of the liquid being weighed, and clogging by dirt is not liable to occur.

3. These advantages are due to the fact that no discharge valve is used in the weighing tank, avoiding the possibility of leakage, and to the reliability of the counterbalancing weight employed, namely, a liquid column of fixed weight.

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* Presented at the Chattanooga meeting (May, 1906) of the American Society Mechanical Engineers, and forming part of Vol. 27 of the *Transactions*.

† For further discussion on this topic consult *Transactions* as follows:

No. 134, vol. 5, p. 63: "Tilting Water Meter for Purposes of Experiment." J. C. Hoadley.

No. 533, vol. 14, p. 676: "On Water Measurements, with Special Reference to the Schinzel Ebonite Water Meter." Friedrich Lux.

No. 708, vol. 18, p. 134: "Calibration of a Worthington Water Meter." John O. Laird.

4. The apparatus accommodates itself to an irregular supply and delivers intermittent charges or units of uniform weight.

5. It consists of two tanks placed one above the other. The upper or receiving tank stores the liquid and automatically delivers it at intervals as required to the lower tank where it is weighed. The lower or weighing tank delivers the unit charge intermittently, the intervals between deliveries varying with the rate of supply.

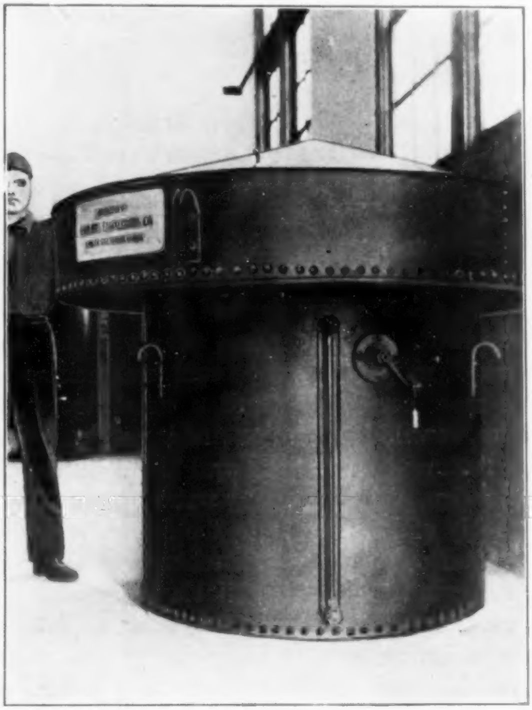


FIG. 1.

6. The general operation of the apparatus is as follows: Liquid accumulates in the upper tank until it overflows a stand pipe located in the upper tank, and spills into the tank below. After a small quantity of overflow has accumulated in the lower tank, the stored liquid in the upper tank is suddenly emptied into the lower tank. Further supply, instead of accumulating in the upper tank as before, flows along its bottom and drops into the lower.

7. The liquid in the lower tank automatically discharges when a unit charge has accumulated, and simultaneously the outlet from the upper tank to the lower tank closes, preventing entrance of liquid into the lower tank while the weighed charge is escaping from it.

8. The apparatus is shown in the accompanying illustrations.

Fig. 1 is from a photograph of the apparatus in use.

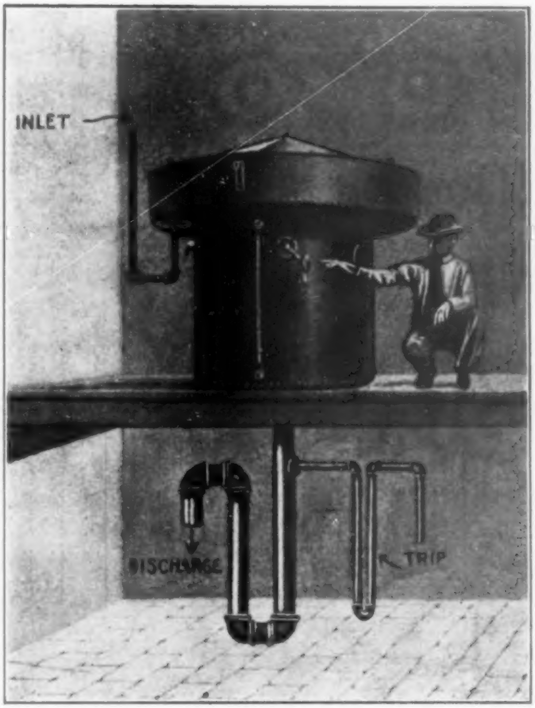


FIG. 2.

Fig. 2 is from a photograph of the same apparatus, showing the trap in the discharge pipe and the "trip" or counterbalancing liquid column by which the unit charge is automatically weighed and discharged.

Fig. 3 is a vertical section of the apparatus adapted for measuring hot water.

Fig. 4 is a detail of the upper tank adapted for measuring cold water.

Fig. 5 is a modified form of trip.

9. Referring to Figs. 3 and 4, (*A*) is the upper or receiving tank, and (*B*) the lower or weighing tank.

(1) Is the inlet for liquid to be weighed, having a baffle plate (1a) to prevent splashing.

(2) Is the valve inlet to the weighing tank. When seated (2) becomes an overflow pipe through which liquid may spill into the weighing tank. When raised, it permits liquid to flow direct into the weighing tank.

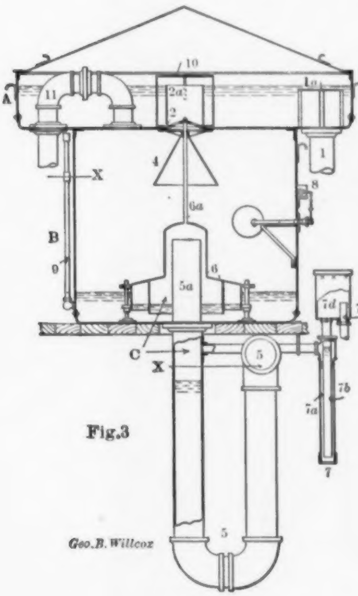


Fig. 3

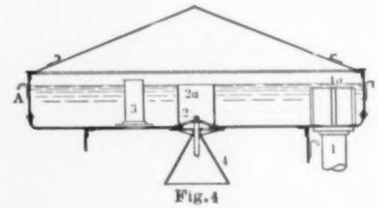


Fig. 4

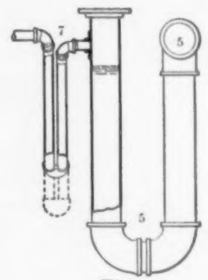


Fig. 5

(3) In Fig. 4 is a vent pipe.

(4) Is a cone to guide the liquid gently to the weighing tank just previous to delivery of the unit charge.

(5) Is a large U-shaped water-sealed discharge pipe, its upper end (5a) projecting through the bottom of the tank (*B*).

(6) Is a vertically movable bell float that envelops the upper end of the pipe (5a). The float and pipe together form a siphon through which the weighed charge escapes when dumping. Float (6) carries a rod (6a), the upper end of which engages and opens

the inlet valve (2) when the float (6) rises. The stored liquid in the upper tank then drops quickly into the weighing tank.

(7) Is a manometer or "trip" by which the charge is automatically weighed and released from tank (*B*).

(8) Is a float-actuated counter that registers the number of charges weighed.

(9) Is a gage glass.

(*C*), Comprising the bell float (6) and the upper part of discharge pipe (5), is an air chamber. This chamber acts as an air cushion while the unit charge is being collected, releasing the unit charge when the air is released by the automatic operation of the trip (7).

When hot water is being weighed two additional devices are employed to prevent steam or vapor entering chamber (*C*) and thereby impairing the accuracy of the apparatus, the first of which devices is an inverted cup (10) carried by the overflow pipe of inlet valve (2). The lower end of the cup (10) is water sealed when the lower tank discharges, thereby preventing hot vapor being drawn from the upper into the lower tank.

(11) Is a pipe connecting the weighing tank and the outside air.

10. The operation is as follows: Assuming the apparatus to be primed, which consists in filling the U-pipe and the trip to their normal levels, liquid from the inlet enters the upper tank, its level rising to the top of the overflow pipe (2a) carried by the valve (2). Overflowing and passing through the open spider of the valve it enters the lower tank. Bell float (6) in the lower tank rises as the charge accumulates, until the shoulder on float rod (6a) engages and lifts valve (2). Thereupon the stored liquid in the upper tank drops quickly into the lower tank, filling it nearly to the discharge level (x). Liquid from pipe (1) thenceforth flows freely along the bottom of the emptied upper tank, gliding down the cone (4) and quietly adding to the level in the lower tank. Soon the level reaches the height (x) for which the tripping device is adjusted and discharge from the lower tank occurs through the automatic operation of the trip (7).

11. The effective head in the manometer or trip (7) exactly adjusts itself to and balances this head of liquid accumulating in the lower tank, increasing as the head in the lower tank increases. The balance is maintained through the medium of the air cushion in chamber (*C*). Releasing the air cushion releases the charge in the lower tank.

12. The function of trip (7) is to suddenly vent the air cushion when the head in the tank equals the maxium head that can be maintained in the trip.

13. When the level in pipe (7a) of the trip sinks to the bottom of the pipe (7b), Fig. 3, the liquid head in the tank is delicately balanced by a liquid column, exactly equal in height to the vertical distance from the bottom of pipe (7b) to the top of pipe (7c). The slightest addition of head in weighing tank (*B*) spills a few drops from (7c) and destroys the hydrostatic balance. Instantly the air cushion in chamber (*C*) shoots the liquid from (7b) and escapes with it through (7c).

14. Deprived of buoyancy, the bell float (6) suddenly drops, allowing the valve (2) to seat, thereby preventing inflow to the weighing tank while the weighed unit discharges. The unit charge (*B*) swiftly siphons out through the bell (6) and pipe (5a), escaping through pipe (5). When the level in the weighing tank has sunk to the rim of bell (6) air enters the bell float, the siphon breaks, and discharge ceases leaving the U-bend of the discharge pipe and the trip full of liquid and the level in the measuring tank at its zero point as in the initial condition. An interval then ensues during which the liquid in the tripping siphon and discharge pipe and the residue in the measuring tank come to rest at normal or zero level, while the next charge begins to accumulate in the upper tank. The process then repeats itself.

15. In the form of trip shown in Fig. 5, the trip is a U-pipe (7a) (7b) capable of swinging around its point of attachment to the discharge pipe. Such adjustment in effect varies the maximum effective head of water column that the pipe may contain and thereby adjusts and controls the discharge level of the measuring tank. The apparatus can thereby be set to discharge any desired weight of unit charge within its capacity.

16. The photograph, Fig. 1, was taken while the apparatus was weighing feed water supplied to a battery of six 350 horse-power Wickes Vertical Water Tube Boilers, generating steam to operate a 1,600 Kilowatt generator with auxiliaries and pumps.

17. The average flow through the tank illustrated is sixty-three thousand pounds of water per hour at a temperature close to the boiling point, fluctuating between 205 degrees and 210 degrees Fahr. It operates under a wide range of supply and delivery, and adapts itself to extraordinary fluctuations.

18. The dimensions of the tank illustrated are as follows: The

upper tank is 72 inches in diameter and 1 foot deep. The lower tank is 48 inches in diameter and $3\frac{1}{2}$ feet deep. Height from floor to top of cover, $5\frac{1}{2}$ feet. The discharge pipe projects below the floor $5\frac{1}{2}$ feet.

19. Numerous trials under actual use as a feed-water weigher showed a maximum error, after four months' constant service, of less than $\frac{1}{4}$ of one per cent.

20. A temperature fluctuation of ten degrees causes no appreciable error in operation. The error that would naturally be introduced by increase of volume incident to increased temperature is small, and even this small error appears in practice to be offset by expansion or contraction of the weighing tank as the temperature rises or falls.

DISCUSSION.

Professor Jacobus.—The statement is made in paragraph 19 of the paper that the machine gives results which are correct within one-quarter of one per cent. I would like to ask whether tests were made with water flowing through the apparatus at various speeds; that is, whether it was tested at full capacity, one-half capacity and one-quarter capacity, and in each case was correct within one-quarter of one per cent. If this was done, and the error does not exceed this small amount, the machine certainly works with remarkable precision.

Mr. Donnelly.—I would like to say a word in commendation of this paper, although I have not had an opportunity to go over the paper very carefully. I do feel confident, however, that there will appear a real demand for such an instrument. Of course, we have water meters without number, but they seem to be rather poorly adapted to keep a record of the amount of water that is evaporated in a boiler; and, as you all know, we do not depend upon the water meter in making a boiler test at all, we are all given to using a method of weighing into barrels, and that is such a cumbersome method and necessitates so much labor and attention that there is no hope that it will ever be used as a permanent device. Owners of boiler plants think that it is strange that we cannot find any better way of keeping a record of the water put into the boilers. A semi-automatic device such as this is, it seems to me, will appeal to users of boilers, and if used will produce a great advance in boiler practice, which is very much to be desired.

*Mr. George B. Willcox.**—The tests referred to in paragraph 19 were made to ascertain if the accuracy of the apparatus decreased after continued use, on account of scale formation or from other causes.

The machine had been measuring feed water for 2,100 horsepower of boilers for several months, the rate of flow of course fluctuating somewhat as the boiler requirements varied. The unit charge at the end of four months was the same as the unit charge when the apparatus was first installed, within one-quarter of one per cent.

To more fully answer Professor Jacobus's question as to the accuracy of the machine when water flows through it at various speeds, I requested the Engineering Department of the University of Michigan, where one of these machines is used as a part of the laboratory equipment, to make some tests. Professor H. C. Anderson, of the Engineering Department, kindly made the tests, and from his report the following results are obtained.

The tested machine was designed for a flow of 40,000 pounds of water per hour, delivering a unit charge of 1760 pounds.

When delivering 13,000 pounds per hour, or one-third capacity, the weight of each unit charge delivered was 1,758 pounds. This corresponds to an error in weight of twelve-hundredths of one per cent.

At 26,000 pounds per hour, or two-thirds capacity, the weight of each unit charge delivered was 1752 pounds, corresponding to an error of forty-one-hundredths of one per cent.

At 46,000 pounds per hour, or working at 16 per cent. above its rated capacity, the weight of a unit charge was 1769 pounds, corresponding to an error of fifty-three-hundredths of one per cent.

The variation in weight in consecutive charges at the same rate of flow was so small that it came far within the range of accuracy of the platform scales upon which the charges were weighed. The above tests were, therefore, made by dumping ten unit charges into a tank set on platform scales and dividing the weight by ten to get the weight of a single charge.

In connection with Mr. Donnelly's remarks, I would like to point out an instance where I have found the use of this apparatus to be of great advantage.

It was required to find out how much salt could be produced

* Author's closure, under the Rules.

in a salt evaporating pan or grainer 150 feet long by 10 feet wide, by 2 feet deep, fitted with coils of steam pipes for evaporating the brine. The salt production per pound of steam condensed was one of the figures to be obtained on the test.

The grainer was constantly enveloped in a cloud of steam, making it impracticable to stay in the grainer room many hours at a time, although it was necessary to run the test continuously for eight days and nights.

A small apparatus of the kind described in this paper was set up to receive the condensation from the grainer pipes. A counter located on the tank kept track of the unit charges delivered, while an electric contact device on the discharge pipe of the apparatus rang a bell located at the other end of the plant where the data man was stationed every time the tank was discharged. When the bell rang the time was noted so that the interval between discharges could be figured, and from this the varying rates of condensation under different conditions of the grainer could not be noted. The test was run continuously for eight days and nights with two men on at a time, one to weigh the salt and the other to measure the condensation water. In addition, this man had time to read eight thermometers and take seven other data observations every thirty minutes.

Without this automatic record of the condensation water, the test would have been a very tiresome and laborious job.

No. 1115.*

A HISTORY OF THE INTRODUCTION OF A SYSTEM OF SHOP MANAGEMENT.†

BY JAMES M. DODGE, PHILADELPHIA, PA.
(Member of the Society and Past President.)

1. After nearly three years' experience introducing into the establishment with which I am connected the system of shop management, identified with the name of Mr. Fred W. Taylor of this Society, I feel that a brief recital of the moving causes which influenced our Company to take up this work would be of interest. I think also it will form a historical recital of the steps and results on broad lines.

2. The works consist of a machine shop, with its usual accompaniment of store-room, tool-room, pattern shop and power plant, together with the required shop, offices, accounting depart-

* Presented at the New York reunion of members of the Society in February, 1906, and by direction of the Committee on Meetings issued as a paper of the Chattanooga Meeting for the information of members and for further discussion Forming part of Volume 27 of the *Transactions.*

† For further references on this topic consult *Transactions* as follows:—
No. 256, vol. 8, p. 630: "A Problem in Profit Sharing." Wm. Kent.
No. 341, vol. 10, p. 600: "Gain Sharing." Henry R. Towne.
No. 459, vol. 12, p. 755: "The Premium Plan of Paying for Labor." F. A. Halsey.
No. 647, vol. 16, p. 856: "A Piece Rate System." Fred. W. Taylor.
No. 928, vol. 23, p. 341: "Bonus System of Rewarding Labor." H. L. Gantt.
No. 965, vol. 24, p. 250: "Gift Proposition for Paying Workmen." Frank Richards.
No. 1001, vol. 24, p. 1302: "Machine Shop Problem." Chas. Day.
No. 1002, vol. 24, p. 1322: "Graphical Daily Balance in Manufacture." H. L. Gantt.
No. 1003, vol. 24, p. 1337: "Shop Management." Fred. W. Taylor.
No. 1010, vol. 25, p. 49: Slide Rules for the Machine Shop as a part of the Taylor System of Management." Carl G. Barth.
No. 1011, vol. 25, p. 63: "Modifying Systems of Management." H. L. Gantt.
No. 1012, vol. 25, p. 68. "Piece Work." Frank Richards.

ments, drawing-room and engineering forces and the selling organization. There is also quite an extensive department devoted to construction and erection in iron and steel. There is no duplicate work done and no package article made or sold as would be the case in large duplicated lots.

3. At the time we first considered the Taylor System, we prided ourselves on having a thoroughly equipped shop, operated by the best methods known to us as respects general management, general accounting and shop accounting. We thought we were decidedly in advance of others in our particular line of business and even of other machine shops. While we felt that we were not intensely progressive, we were also in a satisfied mood, feeling that it would be rather presumptuous for anyone to suggest that our method and general way of doing things could be improved.

4. It was in this frame of mind that we received word of the surprising work done at the shop of the Bethlehem Steel Company with a grade of tool steel to which the names of Taylor-White was attached. I myself made the trip personally to the shop where it was in use and saw tools of this material ripping heavy nickel steel faster than we were in the habit of turning off brass. I also saw under the shadow of a screen over the point of the cutting tool that it cooled with a dull red heat. I found on computation it was turning off a good big chip at a rate of 140 feet a minute and after twenty minutes there was no let up.

It was something of a shock to me to discover that the wonderfully valuable mechanical training I had had and my twenty years of experience would have to be regarded as obsolete from that moment onward.

5. An inspection of my own shop the following day made it apparent that we were hopelessly behind and that it would be necessary for us to rearrange our whole establishment if we were to keep up with the standards that my previous day's experience had forced upon me. This carried with it the sickening feeling that I was going to spend a fortune, was to reduce dividends for several years, was to make an expenditure of a large amount which would give no result in anything to be properly inventoried as an asset, and one-hundred-and-one other financial and mechanical obstacles. To convince my own tool-maker, who like so many other tool-makers was the best in the country, we took down some of his best achievements in tool-making to the Bethlehem shops and the instant failure of our samples alongside of the Taylor-White

product resulted in our negotiating in a few days later for a shop right.

6. Considerable time was spent in getting tools of the right sort for working on cast iron, with the result that we had one lathe and a few tools to fit it which would do from three to four times as much work on cast iron as we had ever been able to do before. This, however, was only the beginning. When we went further the old machine tools had to be either discarded or new ones of special design substituted, or the old tools rebuilt. Electric driving became necessary and finally our machine shop, which had been run most successfully with a 50 horse-power engine, was absorbing over 150 horse-power and calling for more.

Then it became quite evident that the piece-rate would have to be revised. For instance, if 50 pieces could be made per day on a tool, an error in rate either for or against us would be multiplied by fifty, whereas if the same tool could turn out 200 pieces a day our error in rate fixing would be multiplied by 200.

Mr. Taylor's answer to our question was that a scientific time study would be necessary. We were left to accept this because we were following what we regarded as a much quicker and better method which was that of "guess," and we had in our business a number of men who could guess perfectly. Time soon began to show that these wonderful but unscientific guessers were far from infallible, and the guessing was decidedly inaccurate. We were shocked that our perfectly appointed and well-managed tool-room was becoming nervously prostrated and needed "jacking up."

What looks like a simply jacking up process took eighteen months of hard work, but when we were through we were more than satisfied with the expenditure. Increased output reflected glaringly upon the heretofore considered perfect system of storekeeping and accounting. The receiving-room had to be reorganized to fit the store-room. The routing of material through the shop which had been very satisfactory and simple—we were having from six to twenty men remembering hundreds of details—came also to show signs of mental decay. The instruction of our men, the strain of having their lathes speeded, the changes in personnel were all consequences of our first step.

7. The final result was that we called in the man who had been instrumental in getting us into our difficulties and asked him to get us out. The more we worked under the able direction of Mr. Taylor and the assistance of his Mr. Carl G. Barth, also a mem-

ber of the Society, the more we were impressed with the fact that Mr. Taylor in formulating his system had taken good points of management from various sources and had skillfully combined them in a harmonized whole. It took over two years for our organization to surrender fully, and so change our mental attitude that we became really receptive. I mean by this that I found no difficulty at all in having the heads of various departments agree that the introduction of the Taylor System would be most desirable, but in every case it was for everybody else in the establishment but entirely unnecessary for him.

8. I might illustrate a cardinal feature of Mr. Taylor's System by asking you to consider the policy of operating a Fall River steamer with a crew of 200 men, all of whom were in such authority that they were entitled to make suggestions, raise objections and insist on the whole group proceeding with great caution. Obviously the vessel would be in the greatest peril all the time. The one method is to have this entire crew of 200 all functionalized, each man doing his own work under general and specific directions, with a trained pilot steering the boat. If the pilot, for his own glory, insisted upon being illumined so that every one could see him, his usefulness would immediately become impaired. I am fully convinced that the successful perpetuation of a business becomes the more certain the further away we get from the old military idea of having all the brains owned and controlled by one man. We have all seen prosperous concerns come to grief because the person who had the brains and ability to build it up had not been broad-minded enough to see that brains and ability were left behind when he died to conduct the business successfully. In an epigram: "Under the old military system every one was supposed to help the boss. Under the Taylor System the boss is obliged to help and assist the others who are under him." Under this each individual is unconsciously training his successor and working himself out of a job! This "working ourselves out of a job" by the ability and training of a successor makes it possible to promote anyone of the works without a loss of efficiency to the whole. The boss is promoted just as much as anyone else, and his promotion comes to him in the form of perfected organization, releasing him from detail and giving him a greater opportunity to devote his brains and his experience to the development and extension of his business.

9. I have endeavored to make plain that my individual mental

attitude and that of my associates was and is in no way unusual. The whole question resolves itself to this. The high-speed steel called for and made necessary a better system than existed in its entirety in any one machine shop. One shop might have a splendid store system, another an unimpeachable accounting system, another a perfect shipping system, and another a superlative system for routing work. Mr. Taylor's endeavor has been to harmonize the good points of management so as to avoid variations in efficiency with high-grade products and compute valuations in the curve in which we illustrate it. The horizontal line, practically straight, would represent uniform harmony.

10. That improvements will be made in the Taylor System no one can gainsay, so that modifications fitting it to various lines of manufacture may be made. But its underlying principles of efficient planning, task-setting, functional foremanship, which shall not make laborers out of machinists and errand boys of foremen with a full use of the slide-rule in computations, the proper routing of materials through the works, correct record keeping, predetermined shipping dates and other features of the system will have to stand until better means have been tried out. I am convinced that the systematic study of conditions in a manufacturing plant can best be done by the enthusiastic and intelligent outsider. It is absolutely impossible for any man to be thoroughly posted in every detail of the works with which he is connected.

11. The Taylor System is not a method of pay, a specific ruling of account books, nor the use of high-speed steel. It is simply an honest, intelligent effort to arrive at the absolute control in every department, to let tabulated and unimpeachable fact take the place of individual opinion, to develop "team play" to its highest possibility.

In past years numerous instances have come to my notice of machine work having been done more quickly than formerly, but such achievement was rather like the high speed of a hundred-yard dash, or the lowering of a record on the track, interesting, but bringing about no broad spirit of emulation. Under the system to the actual observer, the trained workman with his vastly increased output is working no harder than when his output was much smaller. He is simply working to his best advantage without distraction and with every possible aid that can be rendered him.

The work for him to do is conveniently placed without his

knowing how it got there; the tools with which he is to work are brought to his hands. Finished pieces are removed promptly. By simply following his instructions he finds his pay very much increased and does not suffer undue fatigue, and is relieved of all mental strain and worry. In other words, the man who is the most wonderful and complex machine in the shop is treated with every possible consideration from the viewpoint of increasing his efficiency without harm to himself. Good management without high-speed steel will show handsome returns, but the combination of high-speed steel and the Taylor System, or its equivalent in management, will show the highest possible gain, because of the scientific combination of brain and brawn, which in a shop, as in an individual, represents the highest commercial development.

DISCUSSION.

Mr. James M. Dodge.—In introducing the paper which is to form the basis of our talk this evening, it occurs to me to say that, in my opinion, these evening gatherings are intended to be more informal than our regular semiannual conventions, and it is my wish that you would regard the paper that I have presented more as a talk that I would give you individually, if we were in an office or a railway train and I was telling you my experience.

After Mr. Dodge's presentation, Mr. Charles Wallace Hunt presiding, asked from the chair for discussions or questions upon the paper. Mr. Fred W. Taylor, President of the Society, was also present, although preferring to remain in retirement at the back of the room. He was eventually brought forward to answer questions which came up in the discussion.

Mr. George Hill.—I would like specially to inquire concerning the opinion of either Mr. Dodge or Mr. Taylor, as to the minimum size of shop to which the system referred to can wisely be applied? Is the limit 150 or 100 men when account is taken of the cost of introduction?

Mr. H. L. Binsse.—I would like to ask also if this system is applicable to a wide range of machine-shop practice? Anybody who has been in a locomotive shop must have noticed that the methods are widely different than those in a shop where delicate and accurate machine work is called for.

Mr. Charles B. Rearick.—I am personally interested in a point touched on in the paper which is the system of figuring out deliveries in advance. I speak feelingly because in my own ex-

perience the shop men seem to be unable to arrive at a proper conclusion so as to advise the Selling Department.

Mr. Fred W. Taylor.—Mr. Hill has asked a most pertinent question, if put in the form "In how small a shop can the whole of the mechanism which we approve be applied?" I have in Philadelphia been recently systematizing a shop employing about 120 men, and my present opinion is that this is practically the minimum shop. We were turning out about $10,000 worth of work a month, and that was the limit under the plan formerly in use. For the last three months that shop has been doing a business represented by $25,000 a month instead of $10,000 and at the same time the pay-roll is $300 a week less than it was a year ago. In my opinion, the equipment of that shop could turn out $35,000 worth of work per month, if they had to, with the same force that it had a year ago. On the other hand, if the volume of business in the shop cannot be increased, then I should say that 120 men were too small a number to justify the applying of the system.

Answering Mr. Binsse's question, I would say that the limit in certain departments is from one-thousandth to one-ten-thousandth, and if the same amount of planning be put into accuracy in one place which is put into quantity in another, the result will be as satisfactory. I think that careful study is the keynote, and careful study can bring good results in accuracy as well as in hustling.

A Member.—Suppose the manufacturer has not the demand for the output, could he cut the cost by the application of the Taylor System after it was once in, so that his fifty men, if he had to reduce that number, would still bring him in a profit through saving?

Mr. Taylor.—My judgment would be that if the working force were cut down to fifty men, the cost of the planning, organization, would eat up the profit if the work was complicated. If the work was simple it would not be so.

A Member.—I would like to inquire how Mr. Taylor works out his planning system for a new and difficult piece of work, say the pattern work job which requires ingenuity and thought.

Mr. Taylor.—The gentleman has hit upon the most difficult class of work to be done in any establishment. I used to be a pattern maker myself, and I have never yet attempted to do pattern work under this system. Patterns are necessarily all fresh designs.

Mr. Gus C. Henning.—Is not the system as applicable to pattern

work as any other, since the fundamental idea is to determine the time that it takes to do any particular operation, so that if the work to be done is known, the rate, the time expense and every detail can be determined in advance so that it is all thought and planned before the work is undertaken?

Mr. Taylor.—You are right, but in pattern work it hardly pays to put a special man on the study for one pattern maker until you have done everything else. I can point you, however, to a number of shops where pattern work is done under that plan, although I personally have never tried it.

Mr. F. R. Hutton.—While Mr. Taylor is explaining the details of his system I think it would be illuminating to have him differentiate between the meaning of the word "Order" on the Taylor System and that same word as ordinarily used in shops where his system is not in use.

Mr. Taylor.—This is one of the most important and radical questions which can be asked regarding our system of management. Under the ordinary system and the ordinary training of workmen and foremen, an order from any authority means in a general way "This is what I wished to accomplish; I want that result!" The man who receives that order, if he has anything in him says, "Well, now I have got that order and it is 'up to me' to do a little better if I can. While he told me such and such a thing, he really means 'I want the best that he can do.'"

In the Taylor System with standards adopted through the whole works and the same thing done exactly the same way in a hundred places, it is as bad to do better in one place, from our point of view, as it is to do worse. If one man makes an improvement locally, he throws the other ninety-nine men of the one hundred out of gear as to time, price and the routine of planning. Unless the orders of the man in authority are obeyed the returns are false. On the other hand, we plan a distinct system for improvements with the idea that such improvements shall benefit not alone the one man but the other ninety and nine. It is almost better not to introduce an improved steel for drills in one place, unless you are ready to introduce that faster steel in all places.

Mr. Henshaw.—I would like to ask Mr. Dodge in his establishment what is the method of handling improvements such as Mr. Taylor has spoken of? How does the works get the benefit of an improvement suggested by an employé?

Mr. James M. Dodge.—I called our system "the use of the underground telegraph." We give a man $25 and sometimes $50 for an improvement which he has made and which we have put in use. It makes the men compete with each other in making suggestions. If a man has improvement on his mind and he knows that there is a channel through which it may be developed, he is sure to go there with it. Under the old days improvements were suggested to the foreman; if he was not feeling in good humor the foreman would turn the man down or possibly discharge him for fear that the ability of the suggestion would work him out of his job.

As a matter of fact, the working of the system has been, instead of continually changing the working force, they have become more permanent. The temptation to the foreman to discharge a man because he was endearing himself to the management is no longer present.

Mr. Sanguinetti.—Does the employment of the Taylor System require an appreciable time with new men for them to learn the system?

Mr. Taylor.—We do not get new men all at once, but as they come in the various functional foremen required under our system give such new men more attention than the old ones. Their learning is according to their special ability, some learning faster and others slower.

A Member.—I would like to ask whether Mr. Taylor has experienced any difficulty in applying his system to shops under union control. It happens so often that when improvements are tried a delegation appears and a compromise in the form of a partial backdown is the result. I would like to know Mr. Taylor's experience under these conditions.

Mr. Taylor.—I have never had a strike in my life through the introduction of my system when it was handled right. The shop I have spoken of previously in Philadelphia was completely dominated by unions when we began.

If the steps of an introduction are taken not too fast and in the proper order, going slowly at the start and making no blunders requiring reconsideration, it will not result in driving out the union, but a large number of union men will be converted to the new method. I had several visits from the representative of the Machinists' Union, but as there was nothing doing that any union could take exception to there was no strike ordered and the union

would not have sustained him if he had. We never asked a man to do more work than he was doing before. If the operator objects to increasing the feed from one-sixty-fourth of an inch to one-eighth, he is told that he is to obey orders! If he objects to the shape of a tool, he gets the same answer. There is no issue which any labor union can raise. No labor union will ever step into the shop and say that your men must not take a given feed, if the tool will stand it.

The responsibility for turning out good work does not rest on that man, but on the inspector who comes after. The quality is attended to in exactly the same way as the quantity is by a special man.

The important matter is, first, that the "boss" shall have a carefully and thoroughly laid plan of action and shall follow that just as fast as he can go, so as not to fall down himself or make other people fall down.

At the conclusion of the discussion a vote of thanks, moved by Mr. Rearick, and seconded by many voices, was presented and passed unanimously.

Mr. G. C. Henning.—I think there will be some discussion, Mr. President. For my part, I would like to call attention to the importance of the matter presented. It would be very well indeed to present that paper in detail because it lays before us, for the first time, by a man who has gone through the whole business, the possibilities of economic production in our shops, more of the American practice of the present day, which is ahead of anything the world has ever seen. Nowhere can work be turned out as satisfactorily, quickly and economically, as in the United States at the present time under these methods, with the tools and the materials referred to by Mr. Dodge. It is indeed time that such papers be brought to the notice of all of our manufacturers, because it opens up possibilities never dreamed of before and which at the present day are hardly realized in Europe. The possibilities of cheap production are therein explained, and if the matters be presented to men who have not looked into them thoroughly they will at once understand the results obtained. I think it is well worth while to study a paper of this character because it opens up a field which is going to increase the productivity and economical output of our shops to an extent that is as yet unrealized.

No. 1116.*

COLLAPSING PRESSURES OF BESSEMER STEEL LAP-WELDED TUBES, THREE TO TEN INCHES IN DIAMETER.

BY PROF. REID T. STEWART, PITTSBURG, PA.

(Member of the Society.)

ABSTRACT.

This research was undertaken for the purpose of supplying an urgent demand for reliable information on the behavior of modern wrought tubes when subjected to fluid collapsing pressure. Every means known to engineering science that could aid in the accomplishment of this undertaking has been used, and every possible effort made to get at the truth and have the research yield trustworthy data. It was planned and executed under the immediate direction of the author, at the McKeesport works of the National Tube Company, and has occupied for its completion, during a period of four years, the time of from one to six men.

Series One.—This series of tests was made on tubes that were $8\frac{5}{8}$ inches outside diameter, for all the different commercial thicknesses of wall, and in lengths of $2\frac{1}{2}$, 5, 10, 15 and 20 feet between transverse joints tending to hold the tube to a circular form. The chief purpose of this series of tests was to furnish data for determining which of the existing formulæ, if any, were applicable to modern lap-welded steel tubes, especially when used in comparatively long lengths, such as well casing, boiler tubes and long plain flues.

Series Two.—This series of tests was made on single lengths of 20 feet between end connections, tending to hold the tube to a circular form. Seven sizes, from 3 to 10 inches outside diameter, and in all the commercial thicknesses obtainable, have been tested to date. The chief purpose of these tests was to obtain, for commercial tubes, the manner in which the collapsing pressure of a tube is related to both the diameter and thickness of wall.

* Presented at the Chattanooga, Tennessee, Meeting (May, 1906) of the American Society of Mechanical Engineers and forming part of Volume 27 of the *Transactions*.

Inapplicability of Previously Published Formulæ.—Preparatory to entering upon the present research all existing published formulæ that could be found were collected, and, after the completion of Series One, were tested as to their applicability to modern steel tubes. Among the formulæ thus tested were two each by Fairbairn, Unwin, Wehage and Clark, and one each by Nystrom, Grashof, Love, Belpaire, and the Board of Trade (British), all of which, with possibly two exceptions, appear to be based upon Fairbairn's classical experiments made more than a half century ago, upon tubes wholly unlike the modern product. Without exception, all of these formulæ, when thus tested, proved to be inapplicable to the wide range of conditions found in modern practice. As an illustration of this, the very first tube tested in connection with this research failed under a pressure that exceeded by about 300 per cent. that calculated by means of Fairbairn's formula.

Results of Present Research.—The principal conclusions to be drawn from the results of the present research may be briefly stated as follows:

1. The length of tube, between transverse joints tending to hold it to a circular form, has no practical influence upon the collapsing pressure of a commercial lap-welded steel tube so long as this length is not less than about six diameters of tube. (Pp. 759, 767.)

2. The formulæ, as based upon the present research, for the collapsing pressures of modern lap-welded Bessemer steel tubes, are as follows:

$$P = 1{,}000\left(1 - \sqrt{1 - 1{,}600\,\frac{t^2}{d^2}}\right) \quad . \quad . \quad . \quad . \quad (A)$$

$$P = 86{,}670\,\frac{t}{d} - 1{,}386 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

Where P = collapsing pressure, pounds per sq. inch.
d = outside diameter of tube in inches.
t = thickness of wall in inches.

Formula A is for values of P less than 581 pounds, or for values of $\frac{t}{d}$ less than 0.023, while formula B is for values greater than these.

These formulæ, while strictly correct for tubes that are 20 feet in length between transverse joints tending to hold them to a cir-

cular form, are, at the same time, substantially correct for all lengths greater than about six diameters. They have been tested for seven sizes, ranging from 3 to 10 inches outside diameter, in all obtainable commercial thicknesses of wall, and are known to be correct for this range.

For the convenience of those who wish to apply these formulæ to practice a table has been calculated, giving the collapsing pressures of all the commercial sizes of lap-welded tubes from 2 to 11 inches outside diameter. (See p. 811.)

For those who prefer graphical methods charts have been constructed, for use of which see pp. 816, 819.

When applying these formulæ, tables and charts to practice, it should be remembered that a suitable factor of safety must be applied, which should not be less than from 3 to 6, see p. 815.

3. The apparent fiber stress under which the different tubes failed varied from about 7,000 pounds for the relatively thinnest to 35,000 pounds per square inch for the relatively thickest walls. Since the average yield point of the material was 37,000 and the tensile strength 58,000 pounds per square inch, it would appear that the strength of a tube subjected to a fluid collapsing pressure is not dependent alone upon either the elastic limit or ultimate strength of the material constituting it. (See p. 800.)

Introduction.

The planning and execution of this research was rendered especially difficult because of the lack of any reliable data bearing upon the behavior of modern wrought tubes when subjected to a fluid collapsing pressure. Fairbairn's experiments, made more than a half century ago on tubes unlike the modern product, were of such a character as not to furnish suitable data for the planning of a similar but much more elaborate research on modern tubes. Aside from the numerous formulæ, some ten or twelve in number, based practically upon Fairbairn's experiments, and therefore not to be seriously considered in this connection, the only available data consisted, so far as could be discovered, of a few isolated experiments on flues and several records of the condition under which tubes and flues have failed in service, together with a table of computed collapsing pressures published in a well-known handbook, whose origin could not be traced. As an illustration of the utter unreliability of ex-

isting data on this subject, at the commencement of this research, the very first wrought tube tested failed at a pressure that exceeded by about 300 per cent. that calculated by means of Fairbairn's formulæ.

The experimental part of this research was carried out under the immediate direction of the author at the National Department of the National Tube Company, McKeesport, Pa. He is greatly indebted to the officials of the National Department for the courtesy shown him, especially to the manager, Mr. G. G. Crawford, to the superintendent of the tube mills, Mr. A. M. Saunders, and to Mr. J. A. McCulloch, in whose department the special apparatus was constructed and the experiments conducted. The great interest in the work shown by Mr. McCulloch and his many valuable suggestions as the work progressed were of inestimable value. All the author's wishes in the matter have been cheerfully carried out, the Tube Company generously providing every needful facility for carrying on the research in a most thorough manner.

The exceptional consistency of the results obtained, taking all things into consideration, are due in a large measure to the care with which the author's assistants, Messrs. H. G. Wardale, H. E. Williams and J. N. Kinney, have done their work; and the value of the final conclusions are due largely to Messrs. E. E. Shanor and F. P. Kramer, who have, under his immediate direction, deduced the formulæ representing the results of the experiments and prepared the tables, charts and drawings contained in the body of this paper. It is due Mr. Shanor to state that the greater part of this work has been done by him.

The original Log of Tests comprises, in addition to what has been abstracted for this paper, a complete file of autographic calipering diagrams, photographs showing two views of each tube after being collapsed, impressions from the collapsed sections, and remarks on each individual test. The complete record fills two quarto volumes, each about five inches thick, and, in addition, the matter resulting from working up this data in order to get the final results obtained are sufficient to fill a third volume.

All this matter has been carefully worked over for this paper and condensed into the form of tabulated results and charts showing the consistency of the results obtained, and at the same time revealing to the eye the laws involved.

While much has been necessarily omitted, it is hoped that enough has been given to convince the engineer or artisan, who

may have use for them, of the trustworthiness of the final conclusions.

Preliminary Considerations.

While planning this research it was assumed that the resistance offered by a tube to an external fluid pressure would depend upon the following five things, namely:

1. The diameter of the tube.
2. The length of tube between transverse joints or end connections tending to hold it to a circular form.
3. The thickness of the wall.
4. The deviation of the tube from perfect roundness.
5. The physical properties of the material of which the tube is made.

Of these five things that may vary it was thought that, for the preliminary experiments, at least, Nos. 4 and 5 would be practically constant; No. 4, because the tubes being all made by the same process, would probably run fairly uniform as to deviation from roundness, and No. 5, because the material in this case being Bessemer tube steel, is known to run fairly uniform in its physical properties. The physical tests would, of course, serve as a check upon this latter.

The only variation, then, to be expected in Nos. 4 and 5 would be that due to the inability of the manufacturer to turn out a uniform product. It is recognized here that the physical properties of rolled steel depend in some measure, other things being equal, upon the thickness of the plate; or, in this case, upon the thickness of the wall of the tube. It is clear that any variation of this nature would be a function of the thickness, and would consequently be taken care of in an empirical formula by the quantity representing the thickness.

All the published formulæ bearing upon the subject indicated that the diameter and thickness of wall has each an important determining influence on the collapsing pressure of a tube; and since there were the best of theoretical reasons for believing this to be the case, it was of course decided to plan the research to discover, if possible, the precise nature of this influence over a wide commercial range.

The influence of length of tube, between transverse joints, or end connections tending to hold it to a circular form, upon the collapsing pressure, appeared, in the light of available data, to

be the most uncertain of all the variables entering the problem. It was therefore decided, first of all, to determine the precise nature of this influence.

In order to do this the following apparatus was used, the greater part of which was especially constructed for this research:

Hydraulic Test Apparatus.

The production of a suitable apparatus in which to subject the tubes to an external fluid pressure, and at the same time handle with expedition the large number of tests contemplated, was a somewhat difficult problem to solve. After much consideration of the matter the scheme illustrated in Fig. 1 was adopted.

It will be seen by reference to this figure that the scheme provides for:

1. A test cylinder with one head removable for the reception of the tube to be tested, this cylinder being provided with means for creating an hydraulic pressure within, thus subjecting the tube under test to a fluid-collapsing pressure.

2. A low pressure water supply, *L,* of large volume to rapidly fill the space within the test cylinder not occupied by the tube under test.

3. A variable high pressure water supply, *H,* furnished by an hydraulic pressure pump, *P,* the purpose of which was to create a fluid pressure within the test cylinder, the tube under test by this means being subjected to a gradually increasing fluid-collapsing pressure.

4. A set of pressure gauges, *B, C, D,* having a large range in capacity, connected so that they could be used either singly for indicating the fluid pressure within the test cylinder or in combination for comparison.

5. A vent pipe, *V,* leading from the interior of the tube under test through the head of the test cylinder to the atmosphere, in order to maintain constantly an atmospheric pressure within the tube being tested.

6. An air vent, *E,* connecting with the highest point of the interior of the test cylinder, in order to thoroughly free it from air while being filled with water, after the insertion of a tube to be tested.

In addition to the above, while carrying out this scheme, devices were in use for manipulating the removable head, and for handling the tubes while being entered and withdrawn, but in

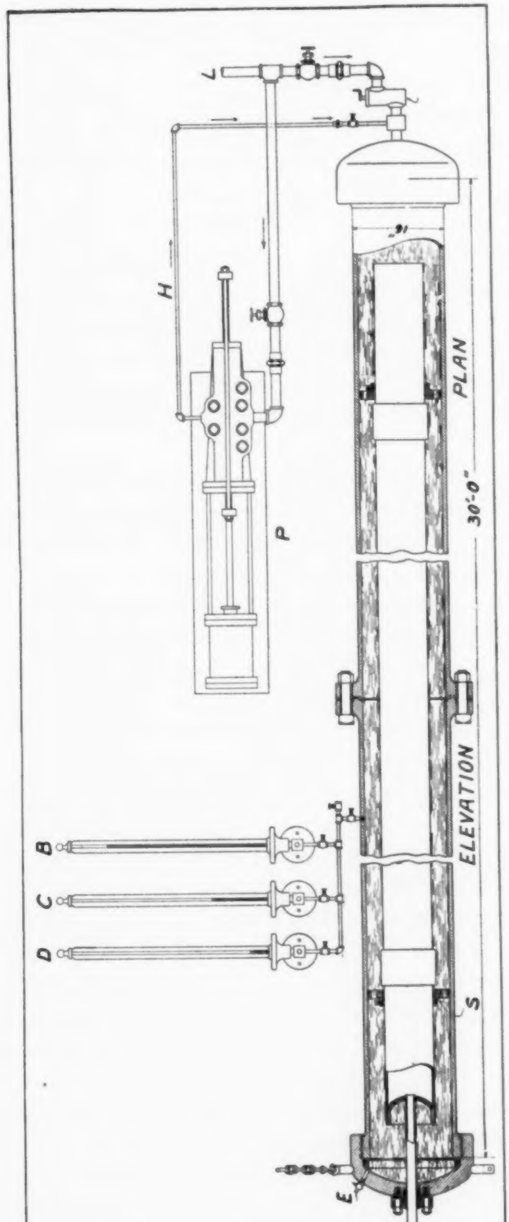


FIG. 1.—SIXTEEN-INCH HYDRALUIC TEST APPARATUS, ESPECIALLY DESIGNED AND CONSTRUCTED FOR COLLAPSING TESTS ON TUBES, CONDUCTED BY PROF. R. T. STEWART.

order not to encumber the paper with unessential details no mention of these will be made.

Sixteen-Inch Test Cylinder.—This cylinder as originally constructed was made up of three sections, whose aggregate length approximated 45 feet, the intention being to have it long enough to accommodate a string of well casing consisting of either two full lengths of 20 feet each, including three couplings, or one full length of 20 feet, with a half length coupled to each of its ends. It was soon discovered that the behavior of a tube in collapse was such that precisely the same results could be had from a single 20-foot length as from either of the above arrangements. Because of this the cylinder was shortened at the first opportunity, by the removal of the intermediate section, to a length of about 30 feet.

The sections of this test cylinder were made from Bessemer steel lap-welded tubes, 16 inches outside diameter and three-quarters inch thick, to which steel flanges were welded for the intermediate joints. Thickening rings were welded to the ends intended to be threaded for the attaching of the heads.

The highest fluid pressure reached in this test cylinder was in connection with the retest of No. 418, which failed under a fluid pressure of 2,890 pounds per square inch. This corresponds to a stress of 28,000 pounds per square inch in the wall of the cylinder. This was about as near the yield point of the material constituting the cylinder as it was thought prudent to go, so that all tests at higher fluid pressures were made in the 8-inch test cylinder, which was relatively about twice as strong.

The heads for the 16-inch test cylinder were made from circular blanks punched from steel plates $2\frac{1}{2}$ inches thick, and pressed into shape, while hot, by means of an hydraulic press. They were then fitted to the ends of the test cylinder, which had been reenforced in the manner already described, by means of trapezoidal threads designed so as to best resist stress in one direction. In the retest on No. 418 these threads were subjected to a shearing stress of 666,000 pounds.

The flange joints connecting the different sections of the test cylinder were tongued and grooved, and were made up with leather packing in the bottom of the grooves. These joints each contained eighteen $1\frac{3}{4}$-inch steel bolts, and were fully as strong against internal fluid pressure as the wall of the cylinder.

Means of Filling Test Cylinder.—The rear end of the test cylinder was connected, in the manner shown in Fig. 1, to the low

pressure water supply, L, of the works, for the purpose of rapidly filling the space within the cylinder and surrounding the tube under test. By this means the test cylinder was quickly filled with water, the pressure within being maintained constantly an atmospheric pressure by means of the air vent, E, shown at the top of the left hand head. This vent also served the purpose of entirely freeing the cylinder from imprisoned air, thus reducing to a minimum the distortion of the tube under test when failure occurred, and also rendering a serious accident to the attendants impossible in case rupture of the cylinder wall should occur while making a test.

Hydraulic Pressure Pump.—The pressure within the test cylinder was created by means of an hydraulic pressure pump capable of working against a fluid pressure up to 3,000 pounds per square inch. Ordinarily this pump was operated, upon entering the region of expected collapse, so as to increase the fluid pressure at a rate of from about 2 to 10 pounds per second, depending upon the gauge used. At these rates of increase of pressure the conditions were favorable for the making of an exact determination of the fluid pressure under which the tube failed.

Pressure Gauges.—The gauges used for indicating the pressure at instant of collapse were three Shaw differential-piston mercury gauges, having capacities of 1,000, 3,000 and 8,500 pounds per square inch. They were connected in the manner shown in Fig. 1, so that, by opening or closing suitable valves, any one or more of them could be connected to the test cylinder for the purpose of indicating the pressure therein. They could also be interconnected for the purpose of comparing their scale readings at different pressures.

The matter of selecting a suitable type of gauge for this research was, at the start, given due consideration. Spring gauges, owing to their liability to become deranged when once calibrated, were not to be considered, and a mercury column for the high pressure expected was out of the question. After considering various forms of dead-weight testers and high-pressure manometers, it was decided to use the Shaw differential-piston mercury gauge. This gauge is in reality a mercury column shortened, for all pressures, to a length of about three feet, by the introduction of differential pistons. These pistons are very ingeniously provided with soft rubber disks, placed so as to render them absolutely fluid-tight, and at the same time practically frictionless. With clean pistons and

new rubber disks these gauges were sensitive and in every respect reliable.

For the service required of them in connection with this research, these gauges were superior to the usual hydraulic spring gauge in three very important respects, namely:

1. The scale of the Shaw mercury gauge as compared with that of the hydraulic spring gauge can be read with about three times the accuracy, that is, the error of scale reading is only about one-third that of the ordinary spring gauge of the same capacity.

2. Since this gauge is in reality a shortened mercury column, it is, when properly constructed, as reliable as the latter. In this respect it bears the same relation to the spring gauge as the mercurial barometer does to the aneroid. It lacks, of course, the closeness with which pressures may be read on a mercury column just in proportion to the relative lengths of their respective scales.

3. The Shaw gauge is practically free from the vibrations that are often so annoying when using a spring gauge. This property of the mercury gauge rendered it eminently serviceable in this connection, since it was necessary to create the fluid pressure by means of a plunger pump without an air cushion.

Eight-Inch Test Cylinder.—This smaller cylinder was constructed for the purpose of testing the 3 and 4-inch tubes, and all of these sizes were tested in it with the exception of Nos. 462 and 464-469. This test cylinder was made up from a single 20-foot length of 8-inch double extra strong pipe, $8\frac{5}{8}$ inches outside diameter, and $\frac{7}{8}$-inch wall.

The details of one end of this cylinder, with tube under test in place, are shown in Fig. 2, the other end being an exact duplicate of the one shown. It will be observed that this apparatus is arranged so as to permit of testing a plain end tube, with the ends open to the atmosphere and the interior of the tube exposed to view while under test. In this way the tube while under test is entirely relieved of any longitudinal stress due to the fluid pressure surrounding it. The sectional view, Fig. 2, shows clearly the construction of the cylinder. It will be observed that the tube is held in place within the test cylinder by steel centering rings, *A*, one at each end, while the cup leather packing rings are being slipped in place over the ends of the tube to be tested. This leather packing ring, at each end of the test cylinder, is backed by a cast iron ring, *B*, that fills the space, as shown, between the inner surface of the end of the test cylinder and the outer surface of the end

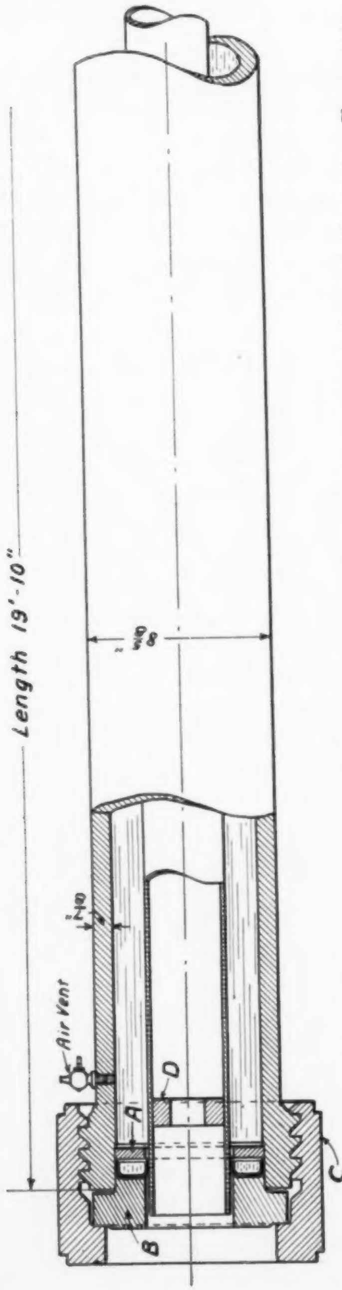


FIG. 2.—EIGHT-INCH HYDRAULIC TEST CYLINDER, SHOWING EXPERIMENTAL TUBE WITH OPEN ENDS IN POSITION. ESPECIALLY DESIGNED AND CONSTRUCTED FOR COLLAPSING TESTS ON TUBES CONDUCTED BY PROF. R. T. STEWART.

of the tube under test. This latter ring is held in place by means of a steel sleeve, *C*, engaging its outer surface by means of an internal flange, and which is attached to the end of the test cylinder by means of the trapezoidal threads shown.

A plug, *D*, was inserted, as shown, near each end of the experimental tubes for the purpose of preventing the centering ring and packing from being damaged and the attendant difficulty of removal of tube that might result from the tube collapsing in the end connections. Since a commercial tube is more apt to collapse at or near one end than near the middle of its length, this simple expedient made it possible to conduct the experiments without the frequent delays that would have otherwise resulted from the jamming of the tube in the end connections.

This smaller test cylinder was placed over and was supported by the larger one. It was connected to the same set of pressure gauges, and was operated, in every essential respect, precisely as was the larger apparatus.

In order to get, with the apparatus available for the purpose, a fluid pressure equal to the greatest working capacity of this cylinder, it became necessary to couple up in series two hydraulic pressure pumps, each of 3,000 pounds capacity, so that the second pump could, if desired, deliver water to the test cylinder under fluid pressures up to 6,000 pounds per square inch. The highest pressure attained in this apparatus was 5,625 pounds per square inch fluid pressure, which was had while testing Nos. 476 and 477.

Test Heads, Supports and Vents.—The different styles of test heads used, the manner of supporting the tube in the test cylinder, and the vent pipes connecting the interior of the tube under test with the atmosphere, are clearly shown in Figs. 3 to 5.

The *Coupled Test Heads*, as shown in Figs. 1 and 3, were made up from short lengths of tubing of the same diameter and thickness of wall as that of the tube placed under test. One end of this test head was threaded like the tube under test, the two being connected by means of a standard sleeve coupling, in precisely the same manner as two sections of the same tubing would be connected in practice, as, for example, in the case of a string of well casing. The other end of each of these test heads was closed by having a steel disk inserted into its end and welded in place, the closed end of the left-hand test head being drilled and tapped for the reception of the end of the vent pipe for maintaining atmospheric pressure within the tube under test, as shown.

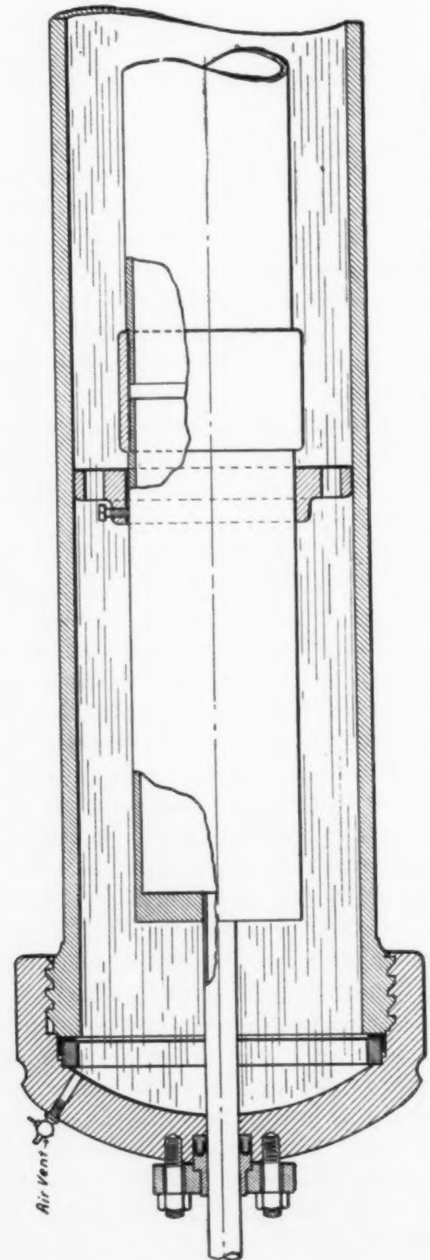


FIG. 3.—EXPERIMENTAL TUBE WITH COUPLED TEST HEAD SHOWN IN POSITION IN 16" HYDRAULIC TEST CYLINDER.
THIS STYLE OF HEAD IS MARKED "*a*" IN THE TABULAR STATEMENTS.

This style of test head was used for all the tests of Series One and for such portions of Series Two as contain the letter "*a*" opposite in column 33 of the tabular statement of principal results of tests.

This method of closing the ends of the experimental tubes, aside from the annoyance of an occasional collapse of the head itself, while entirely satisfactory in other respects, proved to be both slow and expensive to carry out with the facilities at hand for making up the experimental tube before testing, and for the removal of the heads after failure had occurred. Until it was discovered that the influence upon the collapsing pressure due to the tendency of the end connections of a tube to hold it to a circular form, ceased to be measurable, for a commercial tube, at a distance along its length from either end of from 3 to 4 diameters, this style of test head apparently possessed the merit of subjecting the tube under test to the same kind of end support as that actually existing in a string of well casing.

After this fact was fully established the less expensive and otherwise more satisfactory methods below described were used.

The *Bolted Test Head,* Fig. 4, was suggested by the appliance commonly used by tube works for the testing of tubes. In the commercial testing of tubes it is invariably the practice to subject the tube to an internal or bursting pressure; whereas, in connection with this research, an external or collapsing pressure was applied.

This test head (Fig. 4) consisted of a casting with a circular groove cut into its face for the reception of the plain end of the experimental tube. At the bottom of this groove was inserted suitable packing for the production of a water-tight joint when the head is firmly pressed against the end of the tube. The two through bolts shown were intended merely to hold the two heads in place and create sufficient initial pressure to prevent leakage at the start of the test, the external fluid pressure being relied upon, during the continuance of the test, for maintaining a tight joint between the test head and the end of the tube.

These test heads were each provided with two small rollers for the purpose of making easier the handling of the experimental tubes while being inserted and withdrawn from the hydraulic test cylinder. The left-hand head was drilled and tapped, as shown, for the reception of the end of the vent tube. The vent tube for constantly maintaining an atmospheric pressure within the tube under

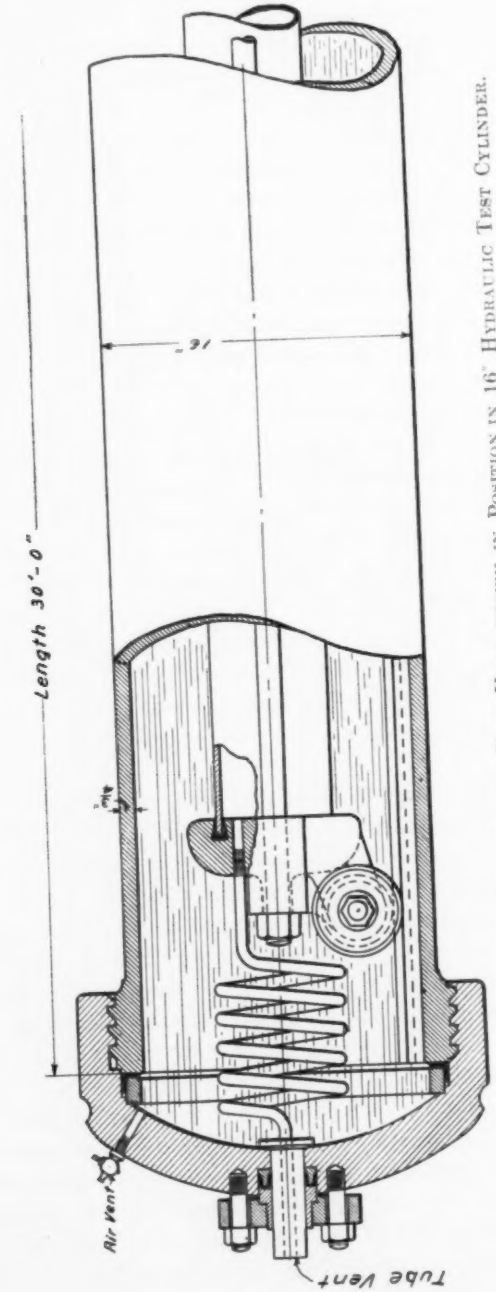


Fig. 4.—Experimental Tube with Bolted Test Head shown in Position in 16" Hydraulic Test Cylinder. This Style of Head is marked "b" in the Tabular Statements.

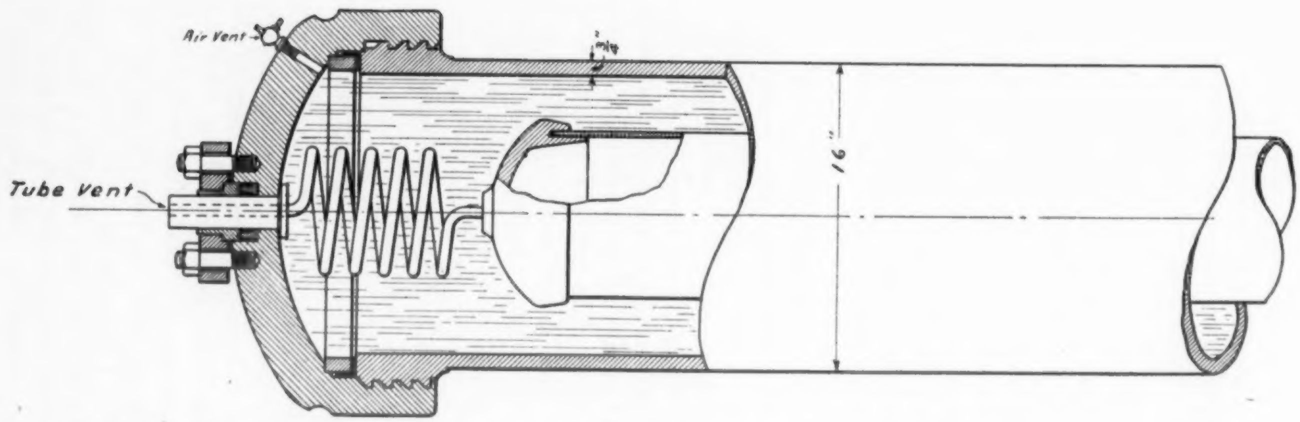


FIG. 5.—EXPERIMENTAL TUBE WITH PRESS-FITTED TEST HEAD SHOWN IN POSITION IN 16" HYDRAULIC TEST CYLINDER. THIS STYLE OF HEAD IS MARKED "c" IN THE TABULAR STATEMENTS.

test, for this style of test head, was coiled into the form of a spiral spring, in order to give it greater flexibility. This style of test head was used for all those experimental tubes of Series Two that are marked "*b*" opposite, in column 33 of the tabular statements.

The *Press Fitted Test Head,* as shown in Fig. 5, was used for the experimental tubes of Series Two that are marked "*c*" opposite in column 33 of the tabular statements. This was the simplest of all the devices tried for closing the ends of the experimental tubes. These heads consisted of iron castings, grooved, as shown, for the reception of the ends of the experimental tubes. These grooves were close-fitting on the sides and contained packing at the bottom, thus forming a very satisfactory joint that became stancher as the external fluid pressure upon the tube under test increased.

The *Vent Pipes* for maintaining constantly an atmospheric pressure within the tube under test, while the external fluid pressure upon it was being gradually increased, are clearly shown in Figs. 3-5. For the coupled test head, where the experimental tube was held central in the hydraulic test cylinder, the vent pipe consisted of a 1½-inch straight pipe, one end of which was screwed into the test head of the experimental tube, while the other end passed through the end of the hydraulic test cylinder to the external atmosphere. A joint stanch against fluid pressure was maintained by means of the cupped leather ring packing shown.

For the other two styles of heads, where the experimental tube was not necessarily kept central in the hydraulic test cylinder, the flexible vent pipe, made by coiling a sufficient length of ½-inch gas pipe into a helical form, was used.

Autographic Calipering Apparatus.

Since it was anticipated that the out-of-roundness of the tube under test would exert a controlling influence on its behavior, it was thought best to devise a piece of apparatus that would indicate this deviation from perfect roundness with accuracy and expedition. A number of schemes for accomplishing this result were worked out. Of these two were constructed and used, known respectively as No. 1 and No. 2.

Autographic Calipering Apparatus No. 2 was used in calipering the bulk of the tubes placed under test, and gave most satisfactory results. It was preceded by Autographic Apparatus No. 1, of

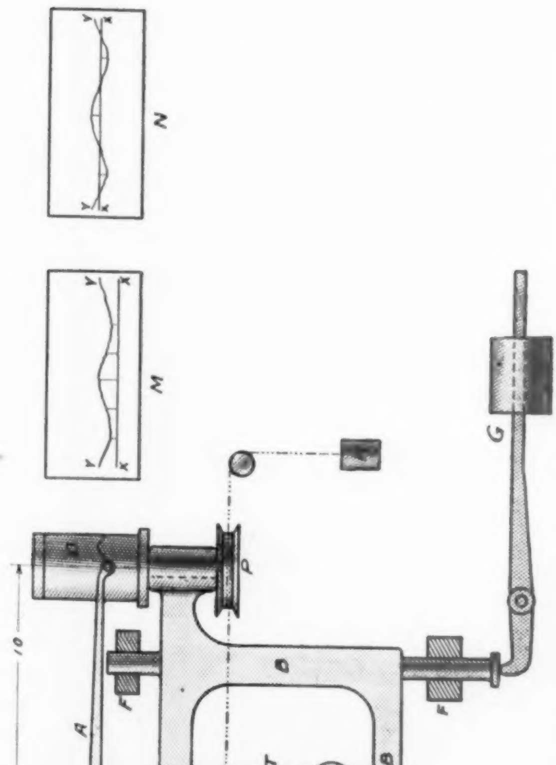


FIG. 6.—SKETCH SHOWING THE PRINCIPLE OF ACTION OF AUTOGRAPHIC CALIPERING APPARATUS No. 2.

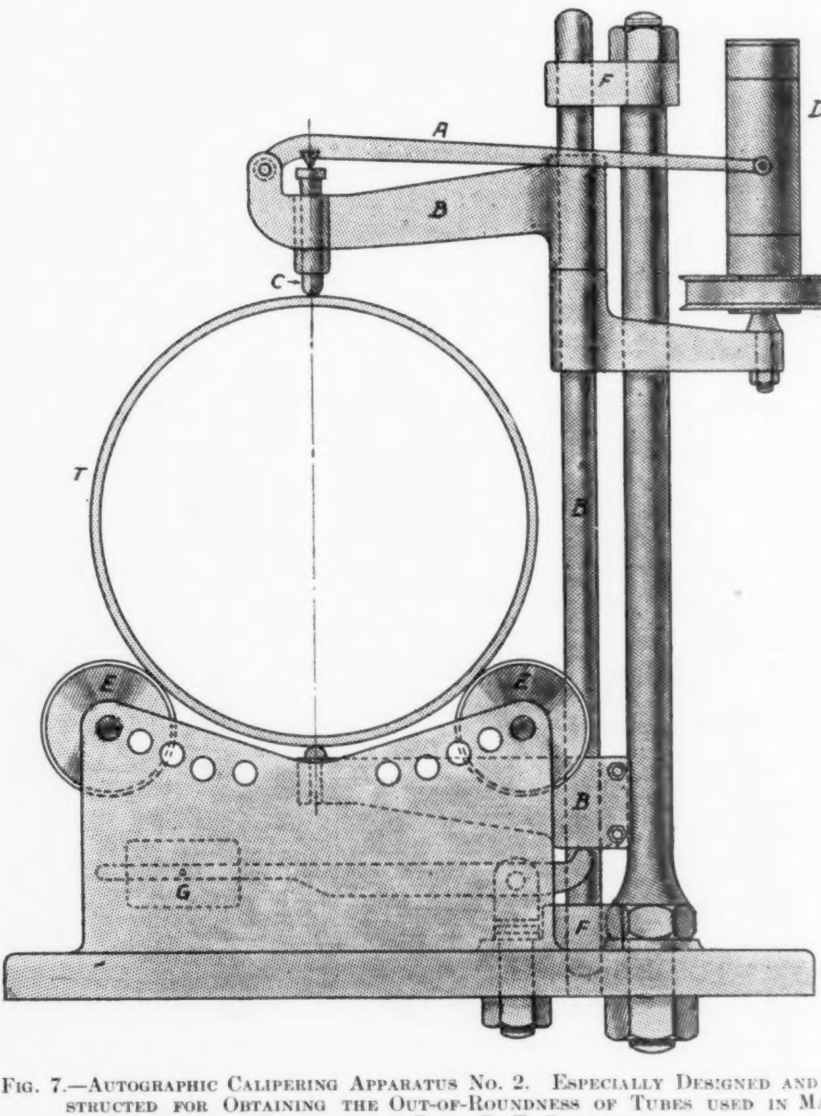


FIG. 7.—AUTOGRAPHIC CALIPERING APPARATUS NO. 2. ESPECIALLY DESIGNED AND CONSTRUCTED FOR OBTAINING THE OUT-OF-ROUNDNESS OF TUBES USED IN MAKING COLLAPSING TESTS CONDUCTED BY PROF. R. T. STEWART.

much lighter weight, constructed on a somewhat different principle. This apparatus proved too flimsy for the service required of it, and was replaced by No. 2 apparatus.

Calipering Apparatus No. 2.—The construction of autographic apparatus No. 2 is clearly shown in Fig. 7, some of the minor details being omitted in order to make the drawing show more clearly the main features of the apparatus; the cord for communicating motion from the tube under test to the recording drum, together with the necessary weights and carrying pulleys, being omitted. These are shown in Fig. 6, which is a diagrammatic form of the apparatus that shows clearly the principle of action.

The tube being calipered is made to rotate by any suitable means on supporting guide wheels, one pair of which are shown at EE. The frame BBB, by means of the guides FF and the counterbalancing lever and weight G, keeps the lower calipering point attached to it constantly in contact with the under surface of the tube while it is being made to rotate. It is evident that any variation in the length of the vertical diameter of the tube while rotating will cause a motion of the upper calipering point C with respect to the frame BBB, which variation is magnified tenfold by the lever A and then recorded on a sheet of paper wrapped about the record drum D. Motion is communicated from the tube T to the record drum D by means of a cord weighted at both ends, in order to prevent slipping, the cord being made to pass once around both the tube and the pulley P attached to the record drum D. In this way the tube being calipered and the record drum are made to rotate synchronously.

To the right are shown two cards taken from the record drum. On these cards the line XX is a reference line, similar to the atmospheric line on an engine indicator card. It is drawn by rotating the record drum by hand while a distance piece is placed between the calipering points, the length of this distance piece being made equal to the nominal outside diameter of the tube being calipered. The result, of course, is a horizontal straight line. The line YY is produced by rotating the tube between the calipering points in the manner described above. The distances between these two lines show, then, to a tenfold scale, the variation of the actual diameters from the nominal diameter for any given cross-section.

Figs. 8-10 show, to a reduced scale, representative examples selected from the numerous autographic records made in connec-

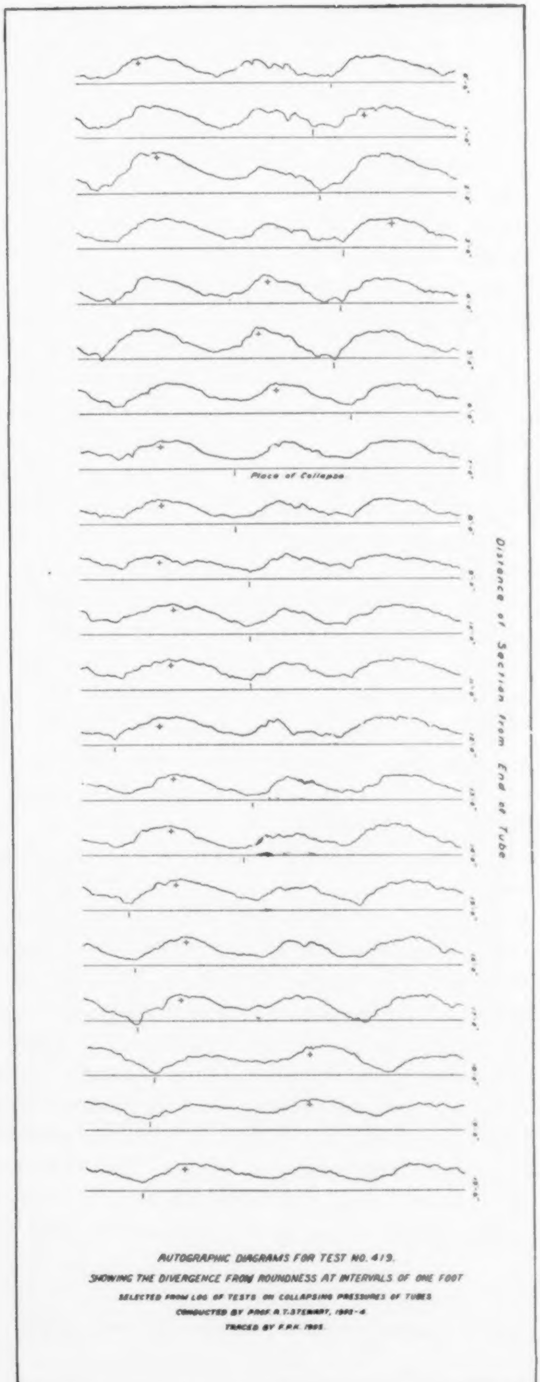
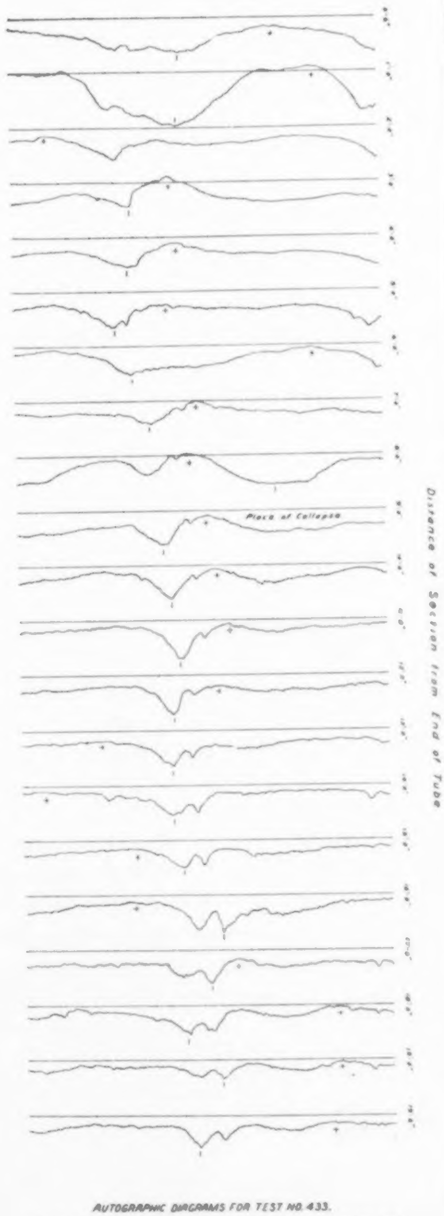

Place of Collapse
Distance of Section from End of Tube
AUTOGRAPHIC DIAGRAMS FOR TEST NO. 419.
SHOWING THE DIVERGENCE FROM ROUNDNESS AT INTERVALS OF ONE FOOT
SELECTED FROM LOG OF TESTS ON COLLAPSING PRESSURES OF TUBES
CONDUCTED BY PROF. R. T. STEWART,
TRACED BY F.P.K.

FIG. 8.



AUTOGRAPHIC DIAGRAMS FOR TEST NO. 433.

SHOWING THE DIVERGENCE FROM ROUNDNESS AT INTERVALS OF ONE FOOT

SELECTED FROM LOG OF TESTS ON COLLAPSING PRESSURES OF TUBES

CONDUCTED BY PROF. R. T. STEWART, 1902–4

TRACED BY F. P. H. 1906

Fig. 9.

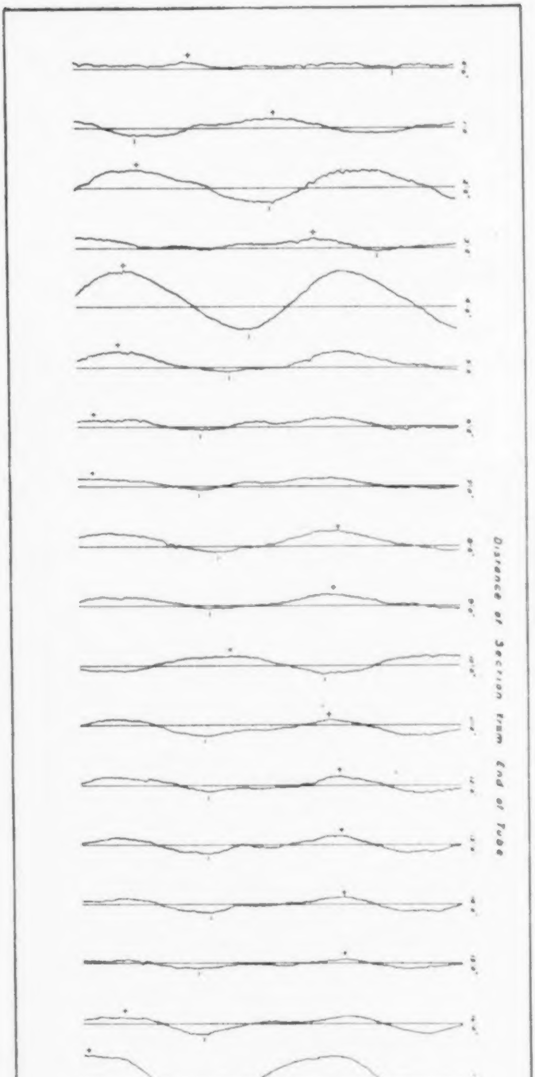

Distance of Section from End of Tube
Place of Collapse
AUTOGRAPHIC DIAGRAMS FOR TEST NO 443
SHOWING THE DIVERGENCE FROM ROUNDNESS AT INTERVALS OF ONE FOOT

FIG. 10.

tion with this research on the collapsing pressures of tubes. Altogether about 6,000 of these autographic records have been made and are on file.

Collapsing Tests, Series One, Showing the Influence of Length of Tube on the Collapsing Pressure.

Since the influence of the length of tube, between transverse joints or end connections tending to hold it to a circular form, upon the collapsing pressure appeared to be the most uncertain element entering the problem, it was thought best, first of all, to determine the precise nature of this influence. Accordingly, it was decided to make a series of tests on a single diameter of tube for all the commercial thicknesses of wall obtainable, in five different lengths of from 2½ to 20 feet.

Selection of Tubes for Testing.—The tubes used in making this series of tests, as well as the other series contained in this paper, were obtained from the National Department of the National Tube Company, McKeesport, Pa., on order issued by the Job Work Shop, in the usual commercial way. Those who filled these orders had no means of knowing for what purpose the tubes were to be used, and presumably, therefore, the tubes thus obtained for purposes of testing represent fair samples of the regular commercial product of the mills.

Every tube thus obtained, without exception, was tested, the complete results of all tests being recorded in the Log, a summary of which appears in this paper. The results may therefore be accepted as indicating the strength to resist fluid collapsing pressures of this Company's Bessemer steel lap-welded tubes, the tubes being taken just as they are found in stock.

Diameter of Tube Tested.—For Series One it was decided to use 8¼-inch well casing, which has a nominal outside diameter of 8⅝ inches. This size was adopted because, taking all things into consideration, it seemed to afford the greatest opportunities for getting at the results desired.

The various diameters of the individual tubes of this series are given in columns 2, 3, 4, 5 and 34 of the tabular statement of principle results of tests, Figs. 11-15. (See folders.)

The *nominal outside diameter*, in inches, appears in column 2, and is for this series 8.625 inches for all tubes tested.

The *average outside diameter,* as made up from measurements on each individual tube, at intervals of one foot along its entire length, are entered in column 3. These measurements were made by means of an especially constructed steel tape, the spacing of whose graduations bore the same relation to those of an ordinary scale divided into inches and hundredths as the length of the circumference of a circle bears to its diameter. That is to say, each inch division on the tape was actually 3.1416 inches long. By this means diameters could be read directly from circumferential measurements, thus making it possible, by means of a single reading, to obtain an average of all the different diameters at any particular foot length of the tube. The advantage of this method will be appreciated when it is remembered that more than 5,000 determinations of mean diameters, at the different cross-sections, one foot apart, had to be made for Series One and Two of this investigation alone, the tubes tested in every case not being perfectly round at any of these sections.

The *greatest and least outside diameters,* at the place of collapse, by which is meant that point of the length of tube where, after failure, the distortion was greatest (see Fig. 16, page 757), are entered respectively in columns 4 and 5.

These entries were made up from the measurements made for out-of-roundness of tube, at each foot along its length, before being placed in the hydraulic test cylinder.

Thickness of Wall.—There were five nominal thicknesses of wall tested in Series One, namely: 0.180, 0.229, 0.271, 0.281, and 0.322-inch, having nominal weights of respectively 16.07, 20.10, 24.38, 25.00 and 28.18 pounds per foot length for the outside diameter of $8\frac{5}{8}$ inches chosen for this series. The actual plain-end weights per foot corresponding to these nominal thicknesses were respectively 16.23, 20.53, 24.18, 25.04, and 28.55 pounds. The nominal thicknesses of wall of the different tubes tested appear in column 6 and the corresponding nominal weights in columns 13 and 34.

The *average thickness* of wall of the tubes of this series appears in column 7. This average thickness for each tube was calculated from the plain-end weight, length, and average outside diameter, as given in column 3. In this way a more exact value could be arrived at for the average thickness than by any other practical means.

SERIES 1 — SHOWING THE INFLUENCE OF LENGTH OF TUBE ON THE COLLAPSING PRESSURE, for lengths of 2½ to 20 feet, between end connections tending to hold the tube to a circular form. For an outside diameter of 8⅝ inches and thicknesses from 0.180 to 0.322 inches

Test Number	Outside Diameter Inches: Nominal	Outside Diameter Inches: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Thickness of Wall Inches: Nominal	Thickness of Wall Inches: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Length of Tube Feet: As Reported	Length of Tube Feet: Actual	Length of Tube Feet: Unsupported	Weight of Tube Lbs per Foot: Nominal	Weight of Tube Lbs per Foot: Actual	Collapsing Pressure: Pounds per Sq Inch	Collapsing Pressure: Gage Used	Collapsing Pressure: Rate of Increase Lbs per Sec	Collaps… Length: In Feet	Collaps… Length: In Dia…
1		8.657	—	—		0.176	0.190	0.170		20.120	19.818		15.92	450		—	6' 0"	8.3
2		8.637	—	—		0.191	0.195	0.179		19.998	19.696		17.24	625		—	6' 3"	8.7
3	8.625	8.641	—	—	0.180	0.186	0.184	0.168	20' 0"	19.997	19.695	16.07	16.77	535	B	—	5' 6"	7.7
4		8.640	—	—		0.183	0.191	0.171		20.004	19.702		16.54	450		—	5' 6"	7.7
5		8.638	—	—		0.191	0.197	0.173		20.011	19.709		17.23	620		—	5' 0"	7.0
Average		8.643				0.185	0.191	0.172		20.026	19.724		16.74	536			5' 8"	7.9
6		8.651	—	—		0.186	0.198	0.184		15.010	14.708		16.82	575		—	6' 0"	8.3
7		8.651	—	—		0.174	0.175	0.145		15.025	14.723		15.76	425		—	6' 0"	8.3
8	8.625	8.655	—	—	0.180	0.183	0.194	0.168	15' 0"	15.004	14.702	16.07	16.545	550	B	—	6' 0"	8.3
9		8.655	—	—		0.191	0.215	0.182		15.010	14.708		17.21	610		—	6' 0"	8.3
10		8.652	—	—		0.188	0.191	0.182		15.002	14.700		16.95	580		—	6' 6"	9.0
Average		8.653				0.184	0.195	0.172		15.010	14.708		16.66	548			6' 1"	8.4
11		8.662	—	—		0.184	0.222	0.182		10.017	9.715		16.65	575		—	5' 0"	7.0
12		8.654	—	—		0.180	0.190	0.174		9.993	9.691		16.26	570		—	5' 0"	7.0
13	8.625	8.666	—	—	0.180	0.177	0.197	0.173	10' 0"	9.997	9.695	16.07	16.03	570	B	—	5' 6"	7.7
14		8.649	—	—		0.171	0.190	0.155		9.997	9.695		15.51	455		—	*9' 0"	*12.5
15		8.651	—	—		0.178	0.197	0.173		10.007	9.705		16.11	550		—	5' 6"	7.7
Average		8.656				0.178	0.199	0.171		10.002	9.700		16.11	548			5' 3"	7.4
16		8.650	—	—		0.182	0.196	0.158		4.990	4.688		16.43	615		—	5' 0"	7.0
17		8.659	—	—		0.181	0.197	0.170		5.010	4.708		16.42	575		—	5' 0"	7.0
18	8.625	8.661	—	—	0.180	0.180	0.191	0.155	5' 0"	5.004	4.702	16.07	16.34	540	B	—	5' 0"	7.0
19		8.665	—	—		0.173	—	—		5.020	4.718		15.64	525		—	5' 0"	7.0
20		8.654	—	—		0.182	0.190	0.178		5.005	4.703		16.43	705		—	5' 0"	7.0
Average		8.658				0.180	0.194	0.165		5.006	4.704		16.25	592			5' 0"	7.0
21		8.658	—	—		0.182	0.188	0.169		2.500	2.198		16.45	915	C	—	2' 6"	3.5
22		8.657	—	—		0.168	0.189	0.177		2.515	2.213		15.23	815	B	14	2' 6"	3.5
23	8.625	8.649	—	—	0.180	0.181	0.185	0.165	2' 6"	2.522	2.220	16.07	16.41	1085	C	—	2' 6"	3.5
24		8.655	—	—		0.170	0.175	0.140		2.514	2.212		15.41	1085	C	—	2' 6"	3.5
25		8.661	—	—		0.181	0.189	0.163		2.507	2.205		16.43	985	C	—	2' 6"	3.5
Average		8.656				0.176	0.185	0.163		2.512	2.210		15.99	977			2' 6"	3.5
26		8.604	8.61	8.58		0.219	0.230	0.210	19' 2"	19.168	18.866		19.57	870	B	—	5' 6"	7.7
27		8.628	8.64	8.61		0.233	0.239	0.186	13' 8"	13.675	13.373		20.95	1115	C	—	5' 6"	7.7
28	8.625	8.659	8.66	8.63	0.229	0.213	0.222	0.195	12' 11"	12.932	12.630	20.10	19.23	850	B	14	5' 6"	7.7
29		8.650	8.69	8.62		0.213	0.221	0.180	12' 8"	12.674	12.372		19.16	750	B	14	5' 6"	7.7
30		8.670	8.69	8.66		0.197	0.211	0.175	12' 3"	12.252	11.950		17.79	650	B	14	5' 6"	7.7
Average		8.642	8.66	8.62		0.215	0.225	0.189					19.34	847		14	5' 6"	7.7
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 11.—TABULAR STATEMENT OF P

R – See remarks on General Remark Sheet

* Not in average, pressure was continued after collapse

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R. T. STEWART, 1902-4 F. P. K., 1905

Collapsed Portion: ...ngth, In Dia's	Collapsed Portion: Distance from End	Collapsed Portion: Angular Distance from Weld	Physical Properties: Tensile Strength Lbs. per Sq. In.	Physical Properties: Yield Point Pounds per Square Inch	Physical Properties: Elongation % in 8 Inches	Physical Properties: Reduction of Area %	Chemical Analysis %: Silicon	Chemical Analysis %: Sulphur	Chemical Analysis %: Phos	Chemical Analysis %: Mang	Chemical Analysis %: Carbon	Chemical Analysis %: Oxide	Material	Remarks	Commercial Designation of Tube as Reported
8.3	11' 0"	− 18°	58 410	35 670	26.13	57.80	.005	.069	.109	.35	.08	—	Bessemer Steel	—	$8\frac{1}{4}$" Casing 16.07 lbs
8.7	15' 6"	− 30°	60 490	38 030	21.04	57.20	.006	.077	.118	.32	.075	—		R	
7.7	4' 3"	+120°	60 020	36 420	24.80	58.73	.006	.077	.115	.31	.075	—		—	
7.7	15' 3"	− 30°	58 640	35 510	21.92	59.43	.006	.079	.104	.32	.08	—		—	
7.8	4' 9"	+ 15°	59 160	36 350	23.00	59.50	.010	.067	.106	.31	.075	—		—	
7.9			59 344	36 396	23.22	58.53	.007	.074	.110	.32	.077				
8.3	4' 0"	− 8°	58 100	37 060	18.54	58.60	.015	.072	.108	.35	.075	—	Bessemer Steel	—	$8\frac{1}{4}$" Casing 16.07 lbs
8.3	9' 0"	− 87°	56 410	35 610	20.75	56.90	.008	.065	.105	.33	.08	—		—	
8.3	3' 6"	− 85°	55 500	36 600	21.46	57.40	.015	.073	.103	.33	.075	—		—	
8.3	4' 0"	−173°	58 520	37 890	21.79	56.70	.008	.073	.112	.33	.07	—		—	
9.0	7' 6"	+ 14°	58 950	36 550	23.21	55.30	.006	.071	.101	.34	.075	—		—	
8.4			57 496	36 742	21.15	56.98	.010	.071	.106	.34	.075				
7.0	3' 6"	+ 27°	57 590	35 340	22.58	60.50	.008	.068	.111	.35	.08	—	Bessemer Steel	—	$8\frac{1}{4}$" Casing 16.07 lbs
7.0	3' 1"	+135°	59 410	38 520	23.75	60.30	.010	.076	.110	.31	.075	—		—	
7.7	3' 10"	+ 17°	55 870	34 200	24.59	60.47	.006	.065	.117	.23	.08	—		—	
*12.5	5' 9"	+ 67°	60 220	37 400	21.67	51.20	—	.079	.110	.38	.085	—		R	
7.7	5' 3"	+ 90°	59 480	38 430	21.88	57.10	.006	.069	.113	.35	.075	—		—	
7.4			58 518	36 778	22.89	57.91	.008	.071	.112	.32	.079				
7.0	2' 6"	+ 32°	—	—	—	—	—	.070	.106	.32	.07	—	Bessemer Steel	—	$8\frac{1}{4}$" Casing 16.07 lbs
7.0	2' 7"	−173°	—	—	—	—	—	.062	.106	.38	.075	—		—	
7.0	2' 8"	+107°	—	—	—	—	—	.076	.117	.28	.07	—		—	
7.0	2' 6"	− 12°	—	—	—	—	—	.088	.113	.37	.08	—		—	
7.0	2' 3"	− 23°	—	—	—	—	—	.074	.112	.34	.07	—		—	
7.0								.072	.111	.34	.073				
3.5	1' 4"	[illegible]	—	—	—	—	—	.085	.121	.26	.076	—	Bessemer Steel	—	$8\frac{1}{4}$" Casing 16.07 lbs
3.5	1' 5"	+135°	—	—	—	—	—	.085	.111	.35	.075	—		R	
3.5	1' 6"	−112°	—	—	—	—	—	.075	.108	.32	.075	—		—	
3.5	1' 3"	0°	—	—	—	—	—	.100	.118	.36	.08	—		—	
3.5	1' 3"	[illegible]	—	—	—	—	—	.075	.112	.32	.07	—		—	
3.5								.084	.114	.32	.075				
7.7	14' 10"	− 58°	56 700	34 060	22.79	59.40	.006	.068	.105	.38	.07	—	Bessemer Steel	R	$8\frac{1}{4}$" Casing 20.10 lbs
7.7	10' 1"	− 9°	57 770	34 730	22.33	53.70	—	.070	.102	.32	.075	—		R	
7.7	8' 2"	− 67°	60 530	37 700	16.67	57.20	—	.083	.117	.35	.08	—		—	
7.7	9' 1"	−145°	61 170	37 300	23.67	59.70	—	.069	.111	.45	.08	—		—	
7.7	4' 2"	− 58°	60 850	37 560	20.71	55.70	.006	.068	.110	.39	.075	—		—	
7.7			59 404	36 270	21.23	57.14		.072	.109	.38	.076				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

...of Principal Results of Tests, Series 1.

SERIES 1 — SHOWING THE INFLUENCE OF LENGTH OF TUBE ON THE COLLAPSING PRESSURE, for lengths of 2½ to 20 feet, between end connections tending to hold the tube to a circular form. For an outside diameter of 8⅝ inches and thicknesses from 0.180 to 0.322 inches

Test Number	Outside Diameter, Inches: Nominal	Outside Diameter: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Thickness of Wall, Inches: Nominal	Thickness: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Length of Tube, Feet: As Reported	Length: Actual	Length: Unsupported	Weight of Tube, Lbs per Foot: Nominal	Weight: Actual	Collapsing Pressure: Pounds per Sq. Inch	Gage Used	Rate of Increase Lbs per Sec.	Collapsed Length: In Feet	Collapsed Length: In Dia's
31		8.658	8.670	8.650		0.210	0.221	0.191	12′ 2″	12.118	11.816		18.96	810		2.2	5′ 6″	7.7
32		8.660	8.690	8.625		0.209	0.235	0.196	11′ 9″	11.753	11.451		18.80	730		2.6	5′ 6″	7.7
33	8.625	8.664	8.670	8.625	0.229	0.226	0.227	0.204	11′ 8″	11.666	11.364	20.10	20.38	840	B	4.0	5′ 6″	7.7
34		8.669	8.690	8.630		0.232	0.254	0.214	11′ 8″	11.668	11.366		20.89	960		2.3	5′ 9″	8.0
Average		8.663	8.680	8.633		0.219	0.234	0.201	11′ 10″	11.801	11.499		19.76	835		2.8	5′ 7″	7.8
35		8.665	8.670	8.660		0.212	0.240	0.211		10.003	9.701		19.17	880		9.9	5′ 6″	7.7
36		8.663	8.670	8.625		0.215	0.240	0.196		10.004	9.702		19.34	842		2.8	5′ 6″	7.7
37	8.625	8.665	8.690	8.630	0.229	0.220	0.258	0.211	10′ 0″	10.003	9.701	20.10	19.82	960	B	1.5	6′ 0″	8.4
38		8.666	8.700	8.630		0.208	0.220	0.187		10.018	9.716		18.77	700		2.5	5′ 0″	7.0
39		8.688	8.730	8.640		0.217	0.232	0.206		10.007	9.705		19.59	845		1.8	5′ 6″	7.7
Average		8.669	8.696	8.637		0.214	0.238	0.202		10.007	9.705		19.34	845		2.1	5′ 6″	7.7
40		8.657	8.660	8.625		0.216	0.241	0.190		5.015	4.713		19.48	975		0.9	5′ 0″	7.0
41		8.653	8.665	8.590		0.208	0.220	0.203		4.988	4.686		18.74	970		2.1	5′ 0″	7.0
42	8.625	8.670	8.700	8.645	0.229	0.210	0.223	0.190	5′ 0″	4.989	4.687	20.10	18.93	805	B	1.7	5′ 0″	7.0
43		8.661	8.690	8.610		0.219	0.193	0.151		4.982	4.680		19.72	875		1.6	5′ 0″	7.0
44		8.664	8.680	8.630		0.207	0.273	0.183		5.013	4.711		18.66	910		2.4	5′ 0″	7.0
Average		8.661	8.679	8.620		0.212	0.230	0.183		4.997	4.695		19.11	907		1.7	5′ 0″	7.0
45		8.650	8.660	8.610		0.210	0.267	0.195		2.490	2.188		18.93	1240		5.0	2′ 6″	3.5
46		8.648	8.650	8.640		0.213	0.226	0.198		2.522	2.220		19.18	1330		3.2	2′ 6″	3.5
47	8.625	8.657	8.670	8.610	0.229	0.214	0.226	0.196	2′ 6″	2.495	2.193	20.10	19.26	1340	C	3.1	2′ 6″	3.5
48		8.677	8.690	8.640		0.210	0.219	0.205		2.508	2.206		19.02	1353		2.6	2′ 6″	3.5
49		8.655	8.690	8.600		0.212	0.220	0.202		2.520	2.218		19.10	1305		4.3	2′ 6″	3.5
Average		8.657	8.672	8.620		0.212	0.232	0.199		2.507	2.205		19.10	1314		3.6	2′ 6″	3.5
50		8.660	8.685	8.585		0.271	0.287	0.265		20.000	19.646		24.28	1435		4.9	5′ 6″	7.7
51		8.688	8.715	8.605		0.274	0.280	0.266		19.988	19.634		24.54	1430		6.2	6′ 0″	8.4
52	8.625	8.664	8.695	8.635	0.271	0.258	0.261	0.248	20′ 0″	20.003	19.649	24.38	23.13	1320	C	3.1	5′ 9″	8.0
53		8.660	8.675	8.625		0.272	0.282	0.255		19.992	19.638		24.32	1520		4.1	5′ 6″	7.7
54		8.660	8.665	8.635		0.262	0.280	0.255		19.993	19.639		23.52	1485		5.2	6′ 0″	8.4
Average		8.666	8.687	8.617		0.267	0.278	0.258		19.995	19.641		23.96	1438		4.7	5′ 9″	8.0
55		8.640	8.645	8.615		0.273	0.299	0.262		14.990	14.636		24.35	1590		5.7	6′ 0″	8.4
56		8.650	8.655	8.615		0.275	0.316	0.262		15.000	14.646		24.55	1420		2.8	6′ 0″	8.4
57	8.625	8.645	8.665	8.635	0.271	0.263	0.268	0.259	15′ 0″	15.010	14.656	24.38	23.54	1590	C	2.2	6′ 6″	9.0
58		8.665	8.685	8.575		0.278	0.290	0.257		15.010	14.656		24.87	1390		5.5	6′ 0″	8.4
59		8.662	8.675	8.635		0.272	0.290	0.260		14.970	14.616		24.33	1710		2.3	6′ 9″	9.4
Average		8.652	8.665	8.615		0.272	0.273	0.260		14.796	14.642		24.33	1540		3.7	6′ 3″	8.7
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 12.—TABULAR STATEMENT OF PRI

R - See remarks on General Remark Sheet

* weld undecided, as marked in photograph should be + 75°

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

...llapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported
...th ... Die's	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq In	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos	Mang	Carbon	Oxide			
7.7	6' 10"	− 34°	57 510	34 450	22.50	57.70	007	057	100	39	08	——	Bessemer Steel	——	8⅝" Casing 20.10 Lbs
7.7	4' 7"	*+ 27°	59 250	37 750	19.17	44.53	——	070	108	25	08	——		——	
7.7	8' 3"	−157°	62 260	38 580	20.67	57.00	006	060	.110	38	075	——		——	
8.0	3' 0"	− 27°	57 330	35 360	23.33	58.40	006	063	102	37	085	——		——	
7.8			59 088	36 535	21.42	54.41	006	063	105	35	08				
7.7	6' 2"	+150°	57 440	34 670	23.75	57.30	——	062	102	31	075	——	Bessemer Steel	——	8⅝" Casing 20.10 Lbs
7.7	7' 0"	−162°	58 870	37 080	23.42	57.30	——	060	097	36	.07	——		R	
8.4	6' 3"	−135°	58 580	41 310	16.46	55.90	——	066	104	31	065	——		——	
7.0	6' 5"	+145°	60 900	38 180	21.33	53.70	——	073	104	29	075	——		——	
7.7	3' 5"	+145°	55 710	34 820	23.13	64.30	005	.043	111	32	08	——		——	
7.7			58 300	37 216	21.62	58.10		061	.104	31	073				
7.0	2' 7"	− 47°	——	——	——	——	——	.071	.114	37	.08	——	Bessemer Steel	R	8⅝" Casing 20.10 Lbs.
7.0	2' 4"	− 80°	——	——	——	——	——	076	.126	.31	075	——		——	
7.0	2' 5"	+135°	——	——	——	——	——	079	.119	.35	085	——		——	
7.0	2' 5"	+ 39°	——	——	——	——	——	.078	107	33	.07	——		——	
7.0	2' 7"	+173°	——	——	——	——	——	.068	100	.36	075	——		——	
7.0								.078	113	.34	077				
3.5	1' 4"	−103°	——	——	——	——	——	070	109	30	07	——	Bessemer Steel	——	8⅝" Casing 20.10 Lbs.
3.5	1' 6"	−157°	——	——	——	——	——	.067	.097	.31	07	——		——	
3.5	1' 4"	+ 21°	——	——	——	——	——	089	.119	.27	07	——		——	
3.5	1' 5"	−160°	——	——	——	——	——	065	.097	.36	07	——		R	
3.5	1' 5"	− 82°	——	——	——	——	——	.066	096	31	065	——		——	
3.5								071	104	31	069				
7.7	16' 10"	+ 22°	58 820	34 920	23.75	57.90	.006	.070	.112	.35	.08	——	Bessemer Steel	——	8⅝" Casing 24.38 Lbs.
8.4	8' 2"	−150°	58 890	37 300	20.92	56.70	——	084	105	35	075	——		——	
8.0	16' 6"	−120°	58 030	37 180	22.79	56.40	005	.072	.111	.32	.07	——		R	
7.7	17' 3"	− 68°	58 650	34 300	24.13	57.80	——	.085	.107	.27	08	——		——	
8.4	7' 5"	−150°	57 500	35 800	20.71	59.50	——	.073	109	.38	.08	——		——	
8.0			58 378	35 900	22.46	57.66		077	.109	.33	.077				
8.4	12' 4"	+160°	58 760	34 920	25.63	57.70	——	.087	.107	.37	.075	——	Bessemer Steel	——	8⅝" Casing 24.38 Lbs
8.4	11' 10"	+132°	57 350	33 900	25.42	59.40	——	.058	.108	.36	08	——		——	
9.0	7' 0"	+130°	56 390	38 000	21.33	51.60	.005	.113	.109	.33	08	——		——	
8.4	7' 9"	+ 10°	57 870	34 420	26.83	58.80	——	.072	.108	.32	075	——		R	
9.4	8' 11"	+ 8°	57 680	36 560	22.58	50.40	006	.078	.111	.39	085	——		——	
8.7			58 010	35 560	24.36	55.58		.082	.109	.35	079				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

...f Principal Results of Tests, Series 1.

SERIES 1 — SHOWING THE INFLUENCE OF LENGTH OF TUBE ON THE COLLAPSING PRESSURE, for lengths of 2½ to 20 feet, between end connections tending to hold the tube to a circular form. For an outside diameter of 8⅝ inches and thicknesses from 0.180 to 0.322 inches

Test Number	Outside Diameter, Inches: Nominal	Outside Diameter: Average	Outside Diameter: At Place of Collapse, Greatest	Outside Diameter: At Place of Collapse, Least	Thickness of Wall, Inches: Nominal	Thickness of Wall: Average	Thickness of Wall: At Place of Collapse, Greatest	Thickness of Wall: At Place of Collapse, Least	Length of Tube, Feet: As Reported	Length of Tube: Actual	Length of Tube: Unsupported	Weight of Tube, Lbs per Foot: Nominal	Weight of Tube: Actual	Collapsing Pressure: Pounds per Sq Inch	Collapsing Pressure: Gage Used	Collapsing Pressure: Rate of Increase Lbs per Sec	Collapsed Length: In Feet	Collapsed Length: In Dia's.
60		8.673	8.685	8.640		0.276	0.290	0.265		9.950	9.596		24.75	1850		4.4	5' 0"	7.0
61		8.668	8.695	8.625		0.265	0.290	0.256		10.010	9.656		23.75	1150		1.6	2' 10"	3.9
62	8.625	8.679	8.695	8.655	0.271	0.268	0.281	0.261	10' 0"	9.990	9.636	24.38	24.05	1628	C	2.9	4' 6"	6.3
63		8.662	8.675	8.660		0.267	0.295	0.259		9.995	9.641		23.94	1585		3.9	7' 0"	9.7
64		8.657	8.675	8.615		0.260	0.283	0.234		10.020	9.666		23.28	1458		1.7	5' 6"	7.7
Average		8.668	8.681	8.639		0.267	0.288	0.255		9.993	9.639		23.95	1533		2.9	5' 0"	6.9
65		8.656	8.665	8.635		0.277	0.325	0.260		4.980	4.626		24.10	1750		3.7	5' 0"	7.0
66		8.653	8.695	8.645		0.268	0.280	0.260		4.995	4.641		23.97	1520		4.7	5' 0"	7.0
67	8.625	8.651	8.665	8.615	0.271	0.269	0.300	0.250	5' 0"	4.995	4.641	24.38	24.07	1640	C	5.2	5' 0"	7.0
68		8.648	8.655	8.615		0.257	0.297	0.233		4.992	4.638		22.99	1345		1.6	5' 0"	7.0
69		8.670	8.675	8.665		0.268	0.294	0.255		5.005	4.651		24.02	1925		4.2	5' 0"	7.0
Average		8.656	8.671	8.635		0.268	0.299	0.252		4.973	4.639		23.97	1636		3.9	5' 0"	7.0
70		8.673	8.690	8.625		0.253	0.266	0.221		2.501	2.147		22.72	1475		3.0	2' 6"	3.5
71		8.650	8.655	8.625		0.271	0.309	0.267		2.500	2.146		24.20	1930		4.3	2' 6"	3.5
72	8.625	8.662	8.665	8.645	0.271	0.273	0.317	0.258	2' 6"	2.480	2.026	24.38	24.47	1880	C	2.7	2' 6"	3.5
73		8.680	8.695	8.630		0.269	0.312	0.252		2.490	2.136		24.15	1850		4.8	2' 6"	3.5
74		8.644	8.620	8.620		0.274	0.273	0.250		2.500	2.146		24.50	1785		2.6	2' 6"	3.5
Average		8.662	8.665	8.629		0.268	0.295	0.250		2.494	2.120		24.01	1784		3.5	2' 6"	3.5
75		8.640	8.675	8.605		0.274	0.282	0.260	20' 0"	20.003	19.649		24.44	1375		4.1	6' 3"	8.7
76		8.663	8.685	8.655		0.272	0.292	0.271	20' 0"	19.988	19.634		24.38	1410		3.6	—	—
77	8.625	8.660	8.665	8.605	0.281	0.274	0.311	0.262	19' 10"	19.855	19.501	25.00	24.47	1275	C	7.3	7' 9"	10.8
78		8.660	8.700	8.595		0.280	0.294	0.274	18' 9"	18.750	18.396		25.08	1250		9.5	7' 0"	9.7
79		8.664	8.670	8.650		0.266	0.280	0.246	18' 4"	18.340	17.986		23.84	1425		8.5	6' 6"	9.0
80		8.656	8.695	8.585		0.266	0.301	0.258		15.000	14.646		23.85	1385		2.7	6' 6"	9.0
81		8.666	8.735	8.595		0.276	0.286	0.270		14.995	14.641		24.73	1305		3.8	6' 6"	9.0
82	8.625	8.662	8.705	8.625	0.281	0.271	0.276	0.256	15' 0"	14.990	14.636	25.00	24.27	1490	C	4.5	6' 3"	8.7
123		8.665	8.695	8.650		0.297	0.305	0.289		14.995	14.641		26.49	1675		5.3	6' 6"	9.0
124		8.644	8.695	8.575		0.263	0.277	0.253		14.995	14.641		23.56	1250		3.5	6' 3"	8.7
Average		8.659	8.705	8.606		0.275	0.289	0.265		14.995	14.641		24.58	1421		4.0	6' 5"	8.9
83		8.669	8.725	8.585		0.270	0.305	0.263		10.012	9.658		24.20	1440		3.5	6' 0"	8.4
84		8.695	8.710	8.670		0.275	0.271	0.247		10.002	9.648		24.72	1500		6.1	6' 6"	9.0
85	8.625	8.664	8.685	8.615	0.281	0.260	0.273	0.240	10' 0"	10.000	9.646	25.00	23.28	1575	C	4.9	6' 3"	8.7
86		8.663	8.675	8.565		0.278	0.286	0.260		9.990	9.636		24.85	1545		4.8	6' 3"	8.7
87		8.664	8.670	8.615		0.272	0.284	0.268		10.010	9.656		24.40	1645		4.1	6' 3"	8.7
Average		8.671	8.697	8.610		0.271	0.292	0.256		10.003	9.649		24.29	1541		4.7	6' 3"	8.7
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 13.—TABULAR STATEMENT OF PRI

R- See Remarks on General Remark Sheet
N- Not Vented

PRINCIPAL RESULTS OF COLLAPSING TESTS ON
NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES
CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K.,1905.

…llapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported.
…th … Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
7.0	7' 6"	+ 5°	58 510	36 730	23.42	53.20	——	.075	.107	.41	.08	——		R	
3.9	8' 7"	−148°	58 980	37 980	23.34	54.60	.006	.075	.113	.36	.07	——		R	
6.3	2' 6"	− 65°	52 800	35 220	13.91	34.50	——	.115	.107	.35	.08	——	Bessemer Steel	R	8⅝" Casing 24.38 Lbs.
9.7	5' 0"	0°	58 720	37 750	22.30	51.10	——	.088	.110	.37	.075	——		——	
7.7	5' 6"	−170°	56 670	34 130	26.76	61.50	——	.065	.105	.42	.075	——		——	
6.9			57 136	36 362	21.99	50.98		.082	.108	.38	.076				
7.0	2' 7"	+ 10°	——	——	——	——	——	.064	.100	.39	.075	——		——	
7.0	3' 1"	+ 50°	——	——	——	——	——	.089	.114	.41	.08	——		R	
7.0	2' 7"	+100°	——	——	——	——	——	.083	.114	.35	.075	——	Bessemer Steel	R	8⅝" Casing 24.38 Lbs.
7.0	2' 7"	− 35°	——	——	——	——	——	.079	.122	.37	.075	——		——	
7.0	2' 6"	−135°	——	——	——	——	——	.122	.113	.30	.07	——		——	
7.0								.087	.113	.36	.075				
3.5	0' 10"	−120°	——	——	——	——	——	.077	.114	.35	.07	——		R	
3.5	1' 0"	− 15°	——	——	——	——	——	.082	.110	.41	.08	——		R	
3.5	1' 8"	[illegible]	——	——	——	——	——	.079	.113	.29	.075	——	Bessemer Steel	R	8⅝" Casing 24.38 Lbs.
3.5	1' 2"	+ 13°	——	——	——	——	——	.077	.112	.36	.075	——		R	
3.5	1' 3"	− 15°	——	——	——	——	——	.094	.101	.31	.07	——		R	
3.5								.082	.110	.34	.074				
8.7	14' 2"	+ 68°	55 680	32 880	24.54	61.00	.010	.061	.109	.35	.08	——	Bess. Steel	R	
—	14' 11"	+ 40°	59 270	35 530	18.63	58.00	——	.065	.106	.40	.08	——	" "	——	
10.8	9' 5"	−120°	43 740	25 240	15.21	28.30	——	.019	.147	Trace	Trace	1.68	Wro't Iron	R	8" Line Pipe 25.00 Lbs.
9.7	9' 3"	+ 92°	45 510	29 520	13.21	25.30	.054	.038	.110	.11	.065	2.58	" "	R	
9.0	15' 3"	−135°	46 430	26 710	15.92	26.80	.062	.040	.109	.11	.065	1.72	" "	R, N	
9.0	10' 4"	−100°	56 310	34 860	23.67	57.50	.006	.064	.106	.33	.075	——		——	
9.0	8' 2"	0°	58 500	34 590	23.50	61.20	——	.061	.110	.41	.08	——		R, N	
8.7	4' 8"	0°	58 610	38 720	21.42	57.80	——	.059	.105	.35	.08	——	Bessemer Steel	R, N	8" Line Pipe 25.00 Lbs.
9.0	4' 2"	+ 35°	58 130	34 560	20.84	58.90	——	.090	.105	.31	.07	——		——	
8.7	3' 7"	+ 15°	57 710	35 300	23.76	56.10	.005	.087	.121	.34	.07	——		——	
8.9			58 252	35 646	22.68	58.70		.072	.109	.35	.075				
8.4	7' 2"	+ 8°	58 510	34 710	24.79	58.20	——	.058	.110	.38	.08	——		——	
9.0	5' 4"	+152°	59 250	36 270	19.67	58.90	——	.065	.103	.30	.065	——		R, N	
8.7	4' 5"	− 20°	58 240	36 400	23.00	59.60	——	.070	.112	.31	.08	——	Bessemer Steel	——	8" Line Pipe 25.00 Lbs.
8.7	6' 8"	−132°	57 340	35 720	21.00	58.40	——	.070	.109	.34	.08	——		——	
8.7	4' 10"	+165°	58 770	33 910	24.58	58.00	——	.071	.123	.36	.075	——		——	
8.7			58 422	35 402	22.61	58.62		.067	.111	.34	.076				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

…f Principal Results of Tests, Series 1.

SERIES 1 { SHOWING THE INFLUENCE OF LENGTH OF TUBE ON THE COLLAPSING PRESSURE, for lengths of $2\frac{1}{2}$ to 20 feet, between end connections tending to hold the tube to a circular form. For an outside diameter of $8\frac{5}{8}$ inches and thicknesses from 0.180 to 0.322 inches.

Test Number	Outside Diameter Inches				Thickness of Wall Inches				Length of Tube Feet			Weight of Tube Lbs per Foot		Collapsing Pressure			Collapsed P		
	Nominal	Average	At Place of Collapse		Nominal	Average	At Place of Collapse		As Reported	Actual	Unsupported	Nominal	Actual	Pounds per Sq. Inch	Gage Used	Rate of Increase Lbs per Sec	Length		Dis f E
			Greatest	Least			Greatest	Least									In Feet	In Dia's.	
88		8.651	8.675	8.610		0.279	0.281	0.258		4.995	4.641		24.93	1615		5.0	5' 0"	7.0	2
89		8.684	8.675	8.655		0.287	0.285	0.272		4.990	4.636		25.75	1685		2.7	5' 0"	7.0	2
90	8.625	8.685	8.705	8.650	0.281	0.284	0.290	0.256	5' 0"	5.000	4.646	25.00	25.45	1700	C	3.7	5' 0"	7.0	2
91		8.664	8.680	8.650		0.275	0.283	0.268		4.998	4.644		24.61	1625		3.4	5' 0"	7.0	2
92		8.674	8.675	8.675		0.273	0.282	0.270		5.000	4.646		24.50	2030		4.9	5' 0"	7.0	2
Average		8.672	8.682	8.648		0.280	0.284	0.265		4.997	4.643		25.05	1731		4.0	5' 0"	7.0	
93		8.657	8.680	8.580		0.259	0.271	0.251		2.510	2.156		23.23	1985		8.8	2' 6"	3.5	1
94		8.638	8.645	8.615		0.279	0.298	0.280		2.500	2.146		24.90	2200		6.7	2' 6"	3.5	1
95	8.625	8.648	8.680	8.585	0.281	0.278	0.320	0.260	2' 6"	2.508	2.154	25.00	24.82	1765	C	6.5	2' 6"	3.5	2
96		8.650	8.690	8.600		0.268	0.280	0.264		2.500	2.146		24.00	1960		7.3	2' 6"	3.5	1
97		8.673	8.720	8.625		0.261	0.303	0.245		2.505	2.151		23.45	1915		8.4	2' 6"	3.5	1
Average		8.653	8.683	8.601		0.269	0.294	0.260		2.505	2.151		24.08	1961		7.5	2' 6"	3.5	
98		8.666	8.680	8.645		0.308	0.312	0.296		19.990	19.636		27.44	1735		5.6	6' 3"	8.7	14
99		8.663	8.700	8.620		0.264	0.287	0.258		20.000	19.646		23.66	1375		5.4	—	—	2
100	8.625	8.646	8.665	8.615	0.322	0.297	0.318	0.300	20' 0"	19.999	19.645	28.177	26.44	1710	C	6.3	7' 0"	9.7	9
101		8.647	8.705	8.565		0.294	0.297	0.252		19.995	19.641		26.27	1650		7.0	6' 0"	8.4	17
102		8.656	8.675	8.620		0.303	0.312	0.286		20.012	19.658		27.00	1960		5.4	5' 9"	8.0	13
103		8.641	8.665	8.605		0.320	0.325	0.306		14.990	14.640		28.38	2000		8.9	6' 0"	8.4	7
104		8.645	8.695	8.600		0.290	0.315	0.289		14.985	14.630		25.88	1725		7.4	6' 3"	8.7	2
105	8.625	8.648	8.750	8.540	0.322	0.297	0.308	0.292	15' 0"	14.995	14.645	28.177	26.46	1520	C	6.3	5' 9"	8.0	2
106		8.672	8.755	8.580		0.308	0.331	0.295		14.985	14.635		27.48	1685		7.1	6' 3"	8.7	5
107		8.666	8.700	8.615		0.324	0.329	0.306		14.980	14.630		28.87	2025		6.8	5' 9"	8.0	12
Average		8.654	8.713	8.588		0.308	0.322	0.298		14.987	14.636		27.42	1791		7.3	6' 0"	8.4	
108		8.652	8.675	8.620		0.325	0.328	0.305		9.990	9.640		28.90	1780		5.7	5' 9"	8.0	7
109		8.652	8.655	8.630		0.291	0.298	0.271		9.990	9.640		26.00	1905		8.4	6' 6"	9.0	6
110	8.625	8.626	8.630	8.570	0.322	0.306	0.314	0.302	10' 0"	9.990	9.640	28.177	27.20	2055	C	3.5	6' 9"	9.4	4
111		8.642	8.665	8.620		0.287	0.314	0.267		9.980	9.630		25.58	1725		3.7	6' 3"	8.7	2
112		8.672	8.675	8.615		0.314	0.327	0.297		9.995	9.645		28.84	1585		4.9	6' 0"	8.4	4
113		8.643	8.665	8.595		0.307	0.314	0.293		4.990	4.640		27.30	1795		4.6	5' 0"	7.0	2
114		8.673	8.685	8.635		0.307	0.326	0.304		4.985	4.635		27.43	2040		5.5	5' 0"	7.0	2
115	8.625	8.658	8.663	8.625	0.322	0.297	—	—	5' 0"	4.995	4.645	28.177	26.48	1980	C	1.75	5' 0"	7.0	2
116		8.663	8.665	8.645		0.321	0.332	0.317		5.000	4.650		28.60	2225		4.4	5' 0"	7.0	2
117		8.634	8.655	8.605		0.299	0.316	0.291		4.995	4.641		26.58	2325		3.6	5' 0"	7.0	2
Average		8.654	8.667	8.621		0.306	0.322	0.301		4.993	4.642		27.28	2073		3.97	5' 0"	7.0	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	

FIG. 14.—TABULAR STATEMENT OF PRINC

R- See Remarks on General Remark Sheet
N- Not Vented

PRINCIPAL RESULTS OF COLLAPSING TESTS ON
NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES
CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

…psed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported.
…ie's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs. per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
0	2' 6"	−145°	—	—	—	—	—	.075	.113	.31	.07	—		N	
0	2' 2"	+140°	—	—	—	—	—	.054	.091	.34	.065	—		—	
0	2' 4"	+105°	—	—	—	—	—	.062	.099	.27	.065	—	Bessemer Steel	—	8" Line Pipe. 25.00 Lbs.
0	2' 4"	+ 60°	—	—	—	—	—	.077	.126	.33	.07	—		—	
0	2' 6"	− 50°	—	—	—	—	—	.066	.108	.35	.075	—		N	
0								.067	.107	.32	.069				
.5	1' 9"	− 95°	—	—	—	—	—	.075	.114	.34	.07	—		—	
.5	1' 3"	− 15°	—	—	—	—	—	.065	.112	.40	.08	—		R	
.5	2' 0"	− 95°	—	—	—	—	—	.092	.115	.35	.07	—	Bessemer Steel	R	8" Line Pipe. 25.00 Lbs.
.5	1' 4"	−110°	—	—	—	—	—	.073	.114	.30	.07	—		R	
.5	1' 3"	[illegible]	—	—	—	—	—	.072	.109	.32	.07	—		—	
.5								.075	.113	.34	.072				
.7	14' 10"	−118°	55 680	28 830	27.83	65.60	—	.020	.022	.40	.14	—	O.H. Steel	—	
—	2' 7"	180°	58 010	33 630	24.46	60.20	—	.059	.113	.35	.075	—	Bess. "	—	
.7	9' 2"	+ 18°	56 350	34 510	23.75	57.60	—	.067	.112	.33	.08	—	" "	N	8" Line Pipe. 28.177 Lbs.
.4	17' 4"	+112°	58 300	35 590	24.54	57.80	—	.074	.105	.35	.07	—	" "	—	
.0	13' 2"	−160°	60 130	37 490	23.67	56.90	—	.065	.102	.40	.08	—	" "	R, N	
.4	7' 7"	− 12°	57 070	36 010	21.42	56.20	.005	.058	.137	.31	.07	—		—	
.7	2' 11"	− 35°	56 780	34 320	23.79	51.70	—	.077	.104	.28	.07	—		R	
.8	2' 3"	+138°	58 910	38 240	21.25	59.30	—	.082	.110	.35	.075	—	Bessemer Steel	N	8" Line Pipe. 28.177 Lbs.
.7	5' 11"	−165°	57 850	36 610	25.54	57.40	—	.074	.105	.34	.07	—		—	
.8	12' 11"	180°	57 380	33 030	26.46	59.50	—	.075	.108	.40	.08	—		—	
.4			57 598	35 642	23.69	57.22		.073	.113	.34	.073				
.0	7' 6"	180°	54 050	27 250	28.67	59.10	—	.026	.015	.42	.12	—	O.H. Steel	—	
.0	6' 1"	+140°	56 820	35 810	23.17	63.10	—	.067	.112	.31	.065	—	Bess. "	R	
.4	4' 0"	−130°	58 520	37 490	25.63	60.50	.005	.061	.117	.30	.07	—	" "	R	8" Line Pipe. 28.177 Lbs.
.7	2' 8"	−165°	59 270	36 250	21.46	53.30	.005	.061	.107	.35	.08	—	" "	—	
.4	4' 9"	− 80°	57 020	28 590	28.04	61.00	—	.029	.029	.58	.13	—	O.H. "	R	
.0	2' 6"	+ 60°	—	—	—	—	—	.072	.116	.28	.07	—		—	
.0	2' 1"	+ 10°	—	—	—	—	—	.056	.090	.38	.07	—		—	
.0	2' 6"	− 5°	—	—	—	—	—	—	—	—	—	—	Bessemer Steel	R	8" Line Pipe. 28.177 Lbs.
.0	2' 9"	− 20°	—	—	—	—	—	.075	.111	.37	.075	—		R	
.0	2' 6"	+ 80°	—	—	—	—	—	.081	.113	.33	.07	—		R	
.0								.071	.108	.34	.071				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

Principal Results of Tests, Series 1.

SERIES 1 { SHOWING THE INFLUENCE OF LENGTH OF TUBE ON THE COLLAPSING PRESSURE. for lengths of 2½ to 20 feet, between end connections tending to hold the tube to a circular form. For an outside diameter of 8⅝ inches and thicknesses from 0.180 to 0.322 inches

Test Number	Outside Diameter Inches				Thickness of Wall Inches				Length of Tube Feet			Weight of Tube Lbs per Foot		Collapsing Pressure			Collapsed [illegible]	
	Nominal	Average	At Place of Collapse		Nominal	Average	At Place of Collapse		As Reported	Actual	Unsupported	Nominal	Actual	Pounds per Sq Inch	Gage Used	Rate of Increase Lbs per Sec	Length	
			Greatest	Least			Greatest	Least									In Feet	In Dia's.
118		8.645	8.670	8.625		0.294	0.309	0.266		2.512	2.158		26.22	2450		2.6	2' 6"	3.5
119		8.641	8.650	8.600		0.322	0.360	0.314		2.500	2.146		28.60	2490		8.7	2' 6"	3.5
120	8.625	8.640	8.650	8.615	0.322	0.323	0.355	0.311	2' 6"	2.505	2.150	28.177	28.64	2390	C	9.5	2' 6"	3.5
121		8.657	8.660	8.640		0.319	0.305	0.276		2.520	2.170		28.43	2290		9.0	2' 6"	3.5
122		8.645	8.660	8.615		0.298	0.307	0.275		2.510	2.160		26.60	2365		10.9	2' 6"	3.5
Average		8.646	8.658	8.619		0.311	0.327	0.288		2.509	2.157		27.70	2397		8.1	2' 6"	3.5

REMARKS

2. Closed head 18" long, vented head 23½" long. A 6'-0" head was first used on the closed end, but collapsed at 600 pounds

14 Pressure was continued after collapse took place resulting in great distortion

22. This is a second specimen ordered to replace the original which was damaged in the first test by cutting off the coupling. In this first test one head gave way at 1030 pounds, the pipe not being collapsed

26 The closed head gave way on the first trial at 780 pounds, per gage B

27 Three 6'-0" long 20" casing heads (A B & C) collapsed at 720, 898 and 995 pounds respectively. For final test an 18" Full Weight head and a short one of 16" casing were used

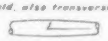
36 Leakage due to bad threads on head

40 Leakage, due to bad threads on head

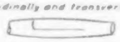
48. Deep transverse grooving by machine at 9" from the end. A defective vent pipe caused leakage on first test

52 Apparently some leakage, not enough to vitiate results

58 Cracked along weld, also transversely



60 Pipe cracked longitudinally and transversely as per sketch



61 See note Nº 52

62 Heads defective on first and second trials. Collapsed on third trial

66 Cracked along bottom of depression

67 Cracked along edge of collapsed portion

REMARKS

70 Leakage too great at 1500 pounds owing to defective heads. Finally collapsed after four trials at 1475 pounds

71 Heads were defective on first trial. Collapsed on second

72 See note Nº 67

73 Collapsed on third trial, leakage too great on first two trials owing to defective heads. Cracked along weld, also in two places along edge of collapsed portion

74 Cracked along the weld

75 Dinge in pipe about 4" in diameter at 10' 9" from end

77 See note Nº 67

78 See note Nº 67

79 See note Nº 66

81 See note Nº 74

82. See note Nº 74

84 See note Nº 71

94 See note Nº 74

95 See note Nº 66

96 Coupling broke off at thread

102 Badly fractured

104. See note Nº 67

109 A slight crack developed

110. A slight report was heard at 2050 pounds, the gages dropped to 2000 pounds, and after about 60 seconds collapse occurred at 2055 pounds. This was an extra good specimen. A small crack developed along the edge of the collapsed portion

REMARKS

112 Cracked along edge of collapsed portion, als [illegible]

115 See note Nº 52

116. Irregular fracture developed. Considerab [illegible] in the head. Four applications of pressure

117 On first test slight report at 2085 pounds, pre [illegible] seconds to 700 pounds. No signs of deformation [illegible] paratus O.K. on test to 2500 pounds. Pipe repla [illegible] 2700 pounds. Filled with water. Pipe collapse [illegible] 5th trial. Pipe cracked in center and broke [illegible]

118 See note Nº 109.

119 This specimen was tested to 2550 pounds [illegible] one of the blank heads gave way. A second [illegible] tion were vitiated by leakage. On a fourth trial [illegible] sed and the vent pipe blown out at 2335 pou [illegible] collapsed as recorded. Pipe collapsed a [illegible] first trial and cracked transversely

120 See note Nº 109

121 Pipe cracked. Large piece blown in altoge [illegible]

122. See note Nº 109

The 1000 lb Gage is marked B
" 3000 lb " " " C
" 8500 lb " " " D

Readings on Gages B and D are re [illegible] on Gage C

FIG. 15.—TABULAR STATEMENT OF PRI [illegible]

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

R – See Remarks on lower part of this sheet

ollapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported
gth In Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
3.5	1' 10"	−140°	——	——	——	——	——	.083	.112	.27	.065	——	Bessemer Steel	R	8" Line Pipe 28.177 Lbs
3.5	1' 1"	+ 20°	——	——	——	——	——	.070	.107	.36	.075	——		R	
3.5	1' 6"	−123°	——	——	——	——	——	.063	.102	.33	.07	——		R	
3.5	1' 3"	0°	——	——	——	——	——	.090	.115	.35	.075	——		R	
3.5	1' 0"	+ 5°	——	——	——	——	——	.088	.121	.34	.07	——		R	
3.5								.079	.111	.33	.071				

RKS

portion, also transversely

Considerable trouble from leaks
essure

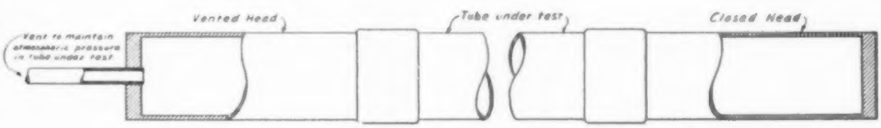
pounds pressure fell in about 15
formation in pipe and testing ap-
Pipe replaced and retested to
e collapsed at 2325 pounds on
and broke at coupling.

0 pounds, and at that pressure
A second and a third applica-
ourth trial one head was collap-
t 2335 pounds. The fifth trial
llapsed at lower pressure than
ely

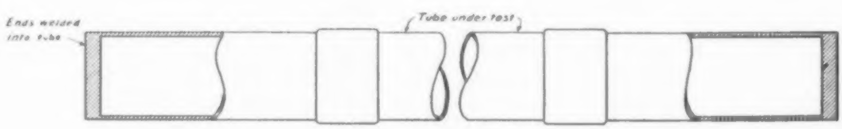
n in altogether detached

ked B
C
D

D are reduced to Readings



This style head used on all no's. of Series 1 except those named below.



This style head used on no's. 79, 81, 82, 84, 89, 92, 100, 102, and 105.

OF PRINCIPAL RESULTS OF TESTS, SERIES 1.

The *greatest and least thickness* of wall at the place of collapse are given respectively in columns 8 and 9. These were obtained from the tube, after collapse, by cutting it across at the point of its length where the distortion appeared to be greatest, after which the greatest and least thicknesses were measured by means of a micrometer caliper.

Weights of Tubes.—The nominal weights, in pounds per foot length, of the tubes tested are given in columns 13 and 34. For this series, having the uniform nominal outside diameter of $8\frac{5}{8}$ inches, these nominal weights were 16.07, 20.10, 24.38, 25.00 and 28.18 pounds per foot length. The actual plain-end weights corresponding to the nominal thicknesses of wall were respectively 16.23, 20.53, 24.18, 25.04, and 28.55 pounds per foot.

The *actual plain end weights* per foot length of the individual tubes tested are entered in column 14. The entries in this column were made up by dividing the weight in pounds of each tube by its length in feet, as given in column 11. The weighing was done on a tested platform scale, and, for those tubes that were threaded before weighing, an allowance was applied for the loss of weight due to threading, this allowance being arrived at experimentally by weighing a number of pieces both before and after threading. In this way the corrections for the different styles of thread were arrived at.

Lengths of Tubes.—The tubes for Series One were ordered in five lengths for each of the five thicknesses tested. These lengths are entered in column 10, and were 20, 15, 10, 5, and $2\frac{1}{2}$ feet, including the threaded ends, for this series. For the tubes of all the other series contained in this report the length ordered in each case was 20 feet, for both plain and threaded ends. It will be noted that the groups Nos. 26 to 34 inclusive, and 77 to 79 inclusive, were supplied in random lengths, presumably, because the stock did not contain tubes of sufficient length, at that date, to fill the order in 20-foot lengths for tubes of these particular weights. All the other tubes were supplied in substantially the lengths as ordered.

The *actual lengths* of the tubes tested, to the nearest thousandth of a foot, as measured by means of a steel tape, are entered in column 11. These measurements for this series include both threaded ends, the coupling that is usually shipped as a part of every threaded tube, and which is ordinarily measured up as a part of its length, not being included in these measurements. The measure-

ments on the tubes of the other series that have threaded ends were made in the same manner.

The *unsupported lengths* of the tubes are entered in column 12. These were arrived at by subtracting the lengths of the portions of the threaded ends that lay inside the couplings from the corresponding actual lengths as given in column 11. Column 12 then shows the actual length of tube exposed to a fluid collapsing pressure, and which, at the same time, received no direct supporting action from any outside source tending to hold it to a circular form. These were the lengths used in deducing the general conclusions from the individual experiments, as shown in Fig. 21.

Collapsing Pressure.—In and near the region of expected collapse the hydraulic pressure within the test cylinder and surrounding the tube under test was increased at the rate, in pounds per second, shown in column 17. This rate in every case was low enough to permit of making accurate readings of the fluid pressure exerted upon the tube under test, and also allow for free elastic deformation of the material constituting the walls of the tubes. In no case was the elastic limit of the material exceeded until after failure had actually occurred. The apparent stress on the wall of the tube at instant of failure ranged from about 7,000 to 34,000 pounds per square inch, respectively, for the relatively thinnest and thickest walls in Series Two. (See page 800 and Fig. 51.)

The fluid collapsing pressure, in pounds per square inch, are entered in column 15, the gauge from which the pressure was read being indicated opposite in column 16, where B, C and D designate respectively the 1,000, 3,000 and 8,500 pounds capacity Shaw differential-piston mercury gauges. These gauges were frequently compared, and the slight differences were adjusted so as to make the readings on B and D conform to those on C, the intention being to have gauge C calibrated after completion of the tests.

Collapsed Portion.—The appearance of the tube after being collapsed in the hydraulic apparatus is clearly shown by the photographs and by the collapsed sections, examples of which are shown in Figs. 16 and 17.

These *photographs of collapsed tubes,* taken in conjunction with the collapse sections, show very clearly the precise nature of the distortion resulting from the subjection of the tube to an external fluid pressure sufficient to cause failure. Referring to Fig. 16, which is a reproduction of the photograph of Nos. 50 to 54 in-

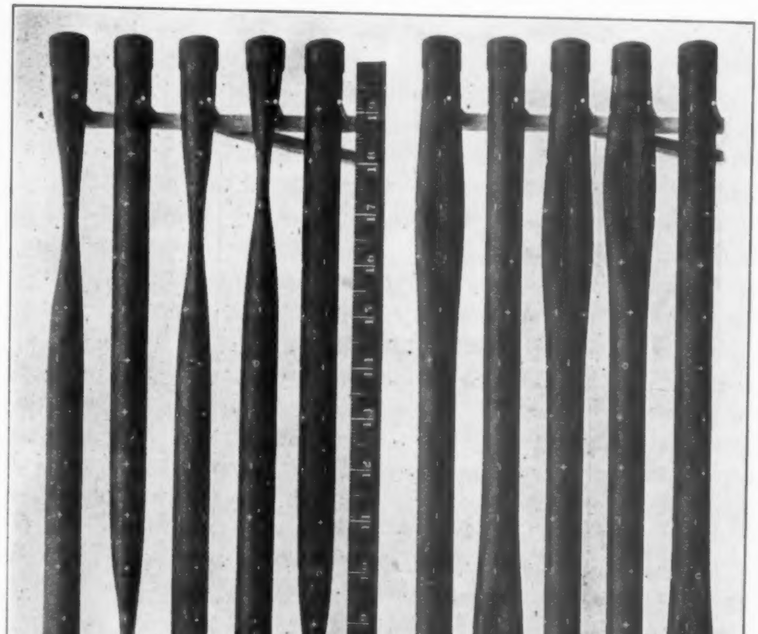


FIG. 16.—PHOTOGRAPH OF TUBES, NOS. 50 TO 54, AFTER BEING COLLAPSED.

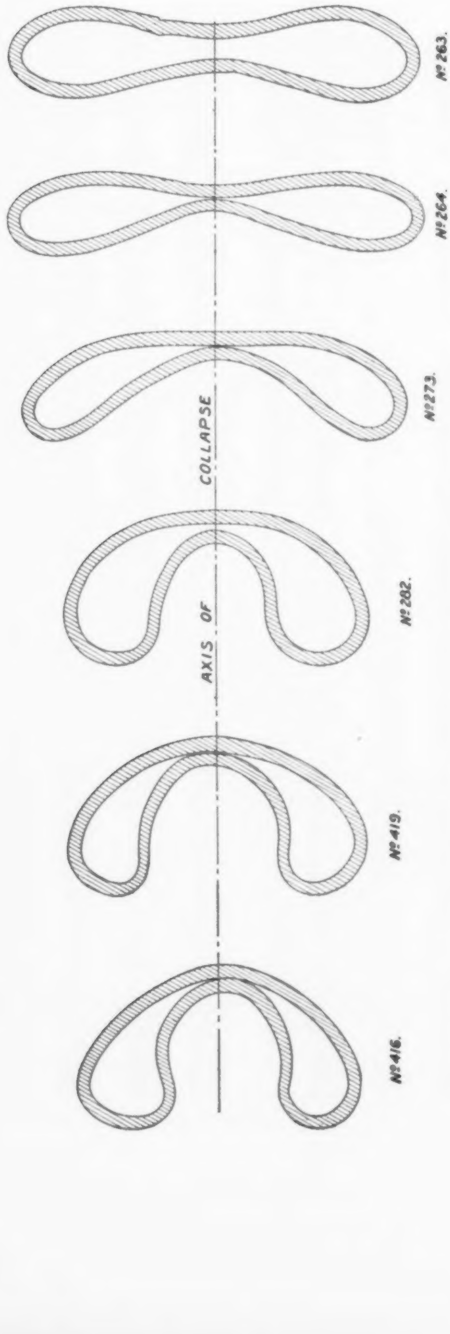


Fig. 17.—Collapse Sections of Tubes from Collapsing Tests on National Tube Co.'s Lap-welded Bessemer Steel Tubes Conducted by Prof. R. T. Stewart.

clusive, it will be observed that two views of each tube are shown. One of these was taken looking in the direction of the axis of collapse, while the other view was taken after the tubes were rotated on their supports through an angle of 90 degrees.

These tubes were all calipered for out-of-roundness, at distances of one foot apart along their entire lengths, before being placed in the hydraulic test apparatus, and, while doing this, the ends of the greatest and least diameters at each of these sections were marked on the tube by means of plus (+) and minus (—) signs, as shown on these photographs. Where noughts (0) appear the tube was so nearly round as to make it difficult to distinguish a greatest and least diameter at that section, even with the exceedingly refined methods in use for making this determination.

The *length of collapsed portion* of tube, in feet, is entered in column 18. This length was determined, after collapse, by measuring the length of the portion of the tube which showed a permanent distortion—for example, referring to Fig. 16, test number 54, it appears that this permanent distortion just becomes measurable at $4\frac{1}{2}$ feet, is greatest at $7\frac{1}{2}$ feet, and terminates at $10\frac{1}{2}$ feet. In this case, then, the length of collapsed portion is 6 feet, which is somewhat less than one-third the length of the tube. More than two-thirds of the length of this tube has suffered no permanent distortion whatever. This localization of collapsed region is typical of all tubes tested in relatively long lengths.

The lengths of the collapsed portions expressed in diameters of the tubes were obtained by dividing the length of collapsed portion of each tube by its outside diameter, both being expressed in inches.

The *distance of collapsed portion from the end of tube,* column 20, was obtained by measuring the distance from the point of greatest distortion—for example, at $7\frac{1}{2}$ feet for test number 54, Fig. 16, to the numbered end of tube, as shown in the photographs. For further discussion of this matter see page 802.

The *angular distance from weld,* column 21, was obtained by measuring the angular distance from the axis of collapse (see Fig. 17) to the weld. Assuming that the observer is stationed at the numbered end of the tube, and looking in the direction of its length, angles measured in the direction in which the hands of a clock rotate are marked plus (+), while those measured in the opposite direction are marked minus (—).

These angular distances were measured by means of an especially constructed tape. On this tape distances were laid off equal to the circumference of the different sizes of tubes to be tested. These distances were then divided into 360 equal parts, each of which would represent the length of one degree of arc of the circumference of the tube. It is evident that a tape constructed after this mannner affords a most satisfactory means for measuring angular distances around tubes. In this way the original angular distance between two points on the surface of a tube, lying in the same transverse plane, can be measured just as readily after the tube has been distorted as before. When it is considered that all of these measurements for angular distance from axis of collapse to the weld had to be made at the place of greatest distortion, after collapse had taken place, the utility of this special tape will be apparent.

Physical Properties of the Steel.—The physical properties entered in columns 22 to 25, inclusive, are the averages from three test specimens cut from each tube, after removal from the hydraulic test apparatus, the test specimens in every case being cut from the undistorted portion, except for the cases, clearly noted in the tabular statement of Series Two, where these specimens were cut from the distorted portion. For these latter it will be observed that the yield point is raised and at the same time the elongation and reduction of area are lowered, more or less according as the portion from which the specimens were cut were more or less distorted. In working up the result for this paper it was assumed that the material of these tubes possessed the same average physical properties as the others of the same series. This seemed but a fair assumption to make under the circumstances, since there was no apparent reason why the material constituting them should differ in respect to physical properties from that of the other tubes tested at the same time.

All the specimens for the physical tests were cut lengthwise of the tube and were pulled in the testing machine without being previously subjected to any straightening action whatever. The test specimens were substantially of the form and dimensions as that adopted for plate metal by the American Section of the International Association for Testing Materials, namely: eight inches between extreme gauge marks, one and one-half inches wide throughout the gauged portion, and enlarged to two inches width at the ends where held in the grip of the testing machine.

The physical tests were made at the National Tube Company's Laboratory, McKeesport, Pa., under the immediate direction of their metallurgist, Mr. G. M. Goodspeed.

Chemical Analysis.—The chemical analyses, columns 26 to 31 inclusive, were also made at the Tube Company's Laboratory, from drillings taken from the tube or from the ends of the physical test specimens.

Material.—The kind of material, whether Bessemer steel, open-hearth steel, or wrought iron, constituting the different experimental tubes, as entered in column 32, was determined for each case by means of the chemical analysis.

It will be observed that the experimental tubes, with but few exceptions, were composed of Bessemer steel. In Series One, three of the tubes tested proved to be wrought iron and also three open-hearth steel, all the others of this series being Bessemer steel. In Series Two, one group of five tubes proved to be wrought iron.

The Bessemer steel constituting the tubes had the following average physical properties:

Tensile strength, pounds per sq. inch	58,000
Yield point	37,000
Elongation in 8 inches, per cent	22
Reduction of area, per cent	57

And the following average chemical analysis:

Sulphur, per cent	.069
Phosphorus, per cent	.106
Manganese, " "	.35
Carbon, " "	.074

Deduction of Law Showing the Relation of Collapsing Pressure to the Length of Tube.

Regrouping of Tests.—In order to arrive at the law expressing the collapsing pressures of Series One in terms of the length of tube and the different thicknesses of wall, for the diameter chosen, it was necessary, first of all, to arrange the tests of this series in the manner shown in the table, Figs. 18-20. In this table all the values entered in the different columns, except the last three of the group of four columns headed "Collapsing Pressure," are taken directly from the corresponding columns of the table of "Principal Results of Collapsing Tests, Series One," Figs. 11-15.

It will be observed that the tests have been regrouped so that,

EFFECT OF LENGTH ON COLLAPSING PRESSURE.— Abstract from Log of Tests Conducted by Prof. R.T. Stewart, 1902-04, on National Tube Co.'s Lap-welded Bessemer Steel Tubes in Lengths of 2½, 5, 10, 15, and 20 Feet, 8⅝" Outside Diameter, to Determine the Effect of Length of Tube on the Collapsing Pressure; to which is Added a Comparison with Values Read from the Curve, M, Representing the Average Results of these Tests. Made by E.E.S. under direction of R.T.S. 1905.

Tests Grouped and Arranged in Order of Length and Thickness of Tube.

Number of Test	Actual Outside Diameter, Inches	Thickness of Wall, Inches: Nominal	Thickness of Wall, Inches: Computed from Wgt.	Unsupported Length of Tube, Feet	Actual Plain End Weight, Lbs. per Ft.	Collapsing Pressure, Pounds per Square Inch: Observed	Corrected to Nom. Thk.	From Curve M	% Variation from M	Designation of Tube, as Reported
22	8.657		0.168	2.213	15.23	815	940		- 3	
24	8.655		0.170	2.212	15.41	1085	1185		+22	2½ Foot Lengths.
23	8.649	0.180	0.181	2.220	16.41	1085	1075	970	+11	
25	8.661		0.181	2.205	16.43	985	965		- 0.5	8⅝" Casing, 16.07 lbs.
21	8.658		0.182	2.198	16.45	915	895		- 7.5	
Average	8.656		0.176	2.210	15.99	977	1012		+ 4.4	
19	8.665		0.173	4.718	15.64	525	605		+ 6	
18	8.661		0.180	4.702	16.34	540	540		- 5	5 Foot Lengths.
17	8.657	0.180	0.181	4.708	16.42	575	565	570	- 1	
16	8.650		0.182	4.688	16.43	615	590		+ 3.5	8⅝" Casing, 16.07 lbs.
20	8.659		0.182	4.703	16.43	705	685		+20	
Average	8.658		0.180	4.704	16.25	592	597		+ 4.7	
14	8.649		0.171	9.695	15.51	455	555		+ 6	
13	8.666		0.177	9.695	16.03	590	625		+19	10 Foot Lengths.
15	8.651	0.180	0.178	9.705	16.11	550	575	525	+ 9.5	
12	8.654		0.180	9.691	16.26	570	570		+ 8.5	8⅝" Casing, 16.07 lbs.
11	8.662		0.184	9.715	16.65	575	535		+ 2	
30	8.670	0.229	0.197	11.950	17.79	650	465	520	-10.5	8⅝" Casing, 20.10 lbs.
Average	8.659		0.181	10.075	16.39	565	554		+ 5.7	
7	8.651		0.174	14.723	15.76	425	490		- 2	
8	8.655		0.183	14.702	16.54	560	520		+ 4	15 Foot Lengths.
6	8.651	0.180	0.186	14.708	16.82	575	510	500	+ 2	
10	8.652		0.188	14.700	16.95	580	495		- 1	8⅝" Casing, 16.07 lbs.
9	8.655		0.191	14.708	17.21	610	495		- 1	
Average	8.653		0.184	14.708	16.66	548	502		+ 0.4	
1	8.657		0.176	19.818	15.92	450	490		+ 3	
4	8.640		0.183	19.702	16.59	450	420		-11.5	20 Foot Lengths.
3	8.641	0.180	0.186	19.695	16.77	535	475	475	0	
2	8.637		0.191	19.696	17.24	625	515		+ 8.5	8⅝" Casing, 16.07 lbs.
5	8.638		0.191	19.709	17.23	620	505		+ 6.5	
Average	8.643		0.185	19.724	16.74	536	481		+ 1.3	
45	8.650		0.210	2.188	18.93	1240	1435		- 1	
48	8.677		0.210	2.206	19.02	1353	1545		+ 6.5	2½ Foot Lengths.
49	8.655	0.229	0.212	2.218	19.10	1305	1480	1450	+ 2	
46	8.648		0.213	2.220	19.18	1330	1495		+ 3	8⅝" Casing, 20.10 lbs.
47	8.657		0.214	2.193	19.26	1340	1495		+ 3	
Average	8.657		0.212	2.205	19.10	1314	1490		+ 2.7	
44	8.664		0.207	4.711	18.66	910	1165		+ 2	
41	8.653		0.208	4.686	18.74	970	1215		+ 6	5 Foot Lengths.
42	8.670	0.229	0.210	4.687	18.93	805	1030	1145	-10	
40	8.657		0.216	4.713	19.48	975	1125		- 2	8⅝" Casing, 20.10 lbs.
43	8.661		0.219	4.680	19.72	875	995		-13	
Average	8.661		0.212	4.695	19.11	907	1106		- 3.4	
38	8.666		0.208	9.716	18.77	700	930	1075	-13.5	
32	8.660		0.209	11.451	18.80	730	950	1055	-10	
31	8.658		0.210	11.816	18.96	810	1020	1050	- 3	
35	8.665		0.212	9.701	19.17	880	1070	1075	- 0.5	10 Foot Lengths.
36	8.663	0.229	0.215	9.702	19.34	842	1000	"	- 7.	
39	8.688		0.217	9.705	19.59	845	980	"	- 9	8⅝" Casing, 20.10 lbs.
37	8.665		0.220	9.701	19.82	960	1060	"	- 1.5	
33	8.664		0.226	11.369	20.38	840	875	1055	-17	
34	8.669		0.232	11.366	20.89	960	925	"	-12.5	
Average	8.666		0.217	10.502	19.52	841	979		- 8.2	
1	2	3	4	5	6	7	8	9	10	11

FIG. 18.

EFFECT OF LENGTH ON COLLAPSING PRESSURE. — **Abstract from Log of Tests Conducted by Prof. R. T. Stewart, 1902-04, on National Tube Co.'s. Lap-welded Bessemer Steel Tubes in Lengths of 2½, 5, 10, 15, and 20 Feet, 8⅝" Outside Diameter, to Determine the Effect of Length of Tube on the Collapsing Pressure; to which is Added a Comparison with Values Read from the Curve, M, Representing the Average Results of these Tests.** Made by E. E. S. under direction of R. T. S., 1905.

Tests Grouped and Arranged in Order of Length and Thickness of Tube.

Number of Test.	Actual Outside Diameter Inches.	Thickness of Wall, Inches.		Unsupported Length of Tube Feet.	Actual Plain End Weight, Lbs. per Ft.	Collapsing Pressure. Pounds per Square Inch.				Designation of Tube, as Reported.
		Nominal	Computed from Wgt.			Observed	Corrected to Nom. Thk.	From Curve M.	% Variation from M.	
28	8.659		0.213	12.630	19.23	850	1015		- 2.5	
29	8.656	0.229	0.213	12.372	19.16	750	915	1040	-12.5	15 Foot Lengths.
27	8.628		0.233	13.373	20.95	1115	1075		- 4	8⅝" Casing, 20.10 lbs.
Average	8.646		0.220	12.792	19.78	905	1002		- 6.3	
26	8.604	0.229	0.219	18.866	19.57	870	970	980	- 1	20 Foot Lengths. 8⅝" Casing, 20.10 lbs.
70	8.673	0.271	0.253	2.147	22.72	1475	1675		-13.5	8⅝" Casing, 24.38 lbs.
93	8.657	0.281	0.259	2.156	23.23	1985	2125		+ 11	8" Line Pipe, 25.00 lbs.
97	8.675	0.281	0.261	2.151	23.45	1915	2030		+ 12	" " " " "
96	8.656	0.281	0.268	2.146	24.00	1940	1975		+ 2	" " " " "
73	8.680	0.271	0.269	2.136	24.15	1850	1870	1935	- 3	8⅝" Casing, 24.38 lbs.
71	8.658	0.271	0.271	2.146	24.20	1930	1930		0	" " " "
72	8.662	0.271	0.273	2.026	24.47	1880	1855		- 5	" " " "
74	8.644	0.271	0.274	2.146	24.50	1785	1750		- 9.5	" " " "
95	8.648	0.281	0.278	2.154	24.82	1765	1685		-13	8" Line Pipe, 25.10 lbs.
94	8.638	0.281	0.279	2.146	24.90	2200	2110		+ 9.3	" " " " "
Average	8.658		0.268	2.135	24.04	1872	1900		- 1.0	2½ Foot Lengths. Corrected to Nom. Thk. 0.271
68	8.648	0.271	0.257	4.638	22.99	1345	1510		- 9	8⅝" Casing, 24.38 lbs.
66	8.653	0.271	0.268	4.641	23.97	1520	1560		- 6	" " " "
69	8.670	0.271	0.268	4.651	24.02	1925	1965		+ 8	" " " "
67	8.651	0.271	0.269	4.641	24.07	1640	1670		+ 0.5	" " " "
92	8.674	0.281	0.273	4.646	24.50	2030	2005	1665	+ 21	8" Line Pipe, 25.10 lbs.
91	8.664	0.281	0.275	4.644	24.61	1625	1570		- 5.5	" " " " "
65	8.656	0.271	0.277	4.626	24.80	1750	1685		+ 1	8⅝" Casing, 24.38 lbs.
88	8.651	0.281	0.279	4.641	24.93	1615	1525		- 8.5	8" Line Pipe, 25.10 lbs.
90	8.685	0.281	0.284	4.646	25.45	1700	1550		- 7	" " " " "
89	8.684	0.281	0.287	4.636	25.75	1685	1495		-10	" " " " "
Average	8.664		0.274	4.641	24.51	1684	1653		- 1.5	5 Foot Lengths. Corrected to Nom. Thk. 0.271
85	8.664	0.281	0.260	9.646	23.28	1575	1695		+ 9.5	8" Line Pipe, 25.10 lbs.
64	8.657	0.271	0.260	9.666	23.28	1450	1570		+ 1.5	8⅝" Casing, 24.38 lbs.
61	8.668	0.271	0.265	9.656	23.75	1150	1215		-21.5	" " " "
63	8.662	0.271	0.267	9.641	23.94	1585	1625		+ 5	" " " "
62	8.679	0.271	0.268	9.636	24.05	1628	1660		+ 7	" " " "
83	8.669	0.281	0.270	9.658	24.20	1440	1450	1550	- 6.5	8" Line Pipe, 25.10 lbs.
87	8.664	0.281	0.272	9.656	24.40	1645	1635		+ 5.5	" " " " "
84	8.695	0.281	0.275	9.648	24.72	1500	1460		- 6	" " " " "
60	8.673	0.271	0.276	9.596	24.75	1850	1795		+ 16	8⅝" Casing, 24.38 lbs.
86	8.663	0.281	0.278	9.636	24.85	1546	1470		- 5	8" Line Pipe, 25.10 lbs.
111	8.642	0.322	0.287	9.638	25.58	1725	1550		0	" " " " "
109	8.652	0.322	0.291	9.640	26.00	1905	1685		+ 9	" " " " "
Average	8.666		0.272	9.642	24.40	1583	1567		+ 1.2	10 Foot Lengths. Corrected to Nom. Thk. 0.271
57	8.646	0.271	0.263	14.656	23.54	1590	1675		+ 13.5	8⅝" Casing, 24.38 lbs.
124	8.644	0.281	0.263	14.641	23.56	1250	1335		- 9.5	8" Line Pipe, 25.10 lbs.
*79	8.664	0.281	0.266	17.986	23.84	1425	1480	—	—	" " " " "
80	8.656	0.281	0.266	14.646	23.85	1385	1440		- 2.5	" " " " "
82	8.662	0.281	0.271	14.636	24.27	1490	1490		+ 1	" " " " "
59	8.662	0.271	0.272	14.616	24.33	1710	1700	1475	+ 15	8⅝" Casing, 24.38 lbs.
55	8.640	0.271	0.273	14.636	24.35	1590	1565		+ 6	" " " "
56	8.650	0.271	0.275	14.646	24.55	1420	1380		- 6.5	" " " "
81	8.666	0.281	0.276	14.641	24.73	1305	1255		-15	8" Line Pipe, 25.10 lbs.
58	8.665	0.271	0.278	14.656	24.87	1390	1315		-11	8⅝" Casing, 24.38 lbs.
104	8.645	0.322	0.290	14.630	25.88	1725	1525		+ 3.5	8" Line Pipe, 28.18 lbs.
Average	8.653		0.273	14.690	24.39	1485	1468		- 6.5	15' Lengths, to Nom. Thk. 0.271 * See 20 Foot Lengths.
1	2	3	4	5	6	7	8	9	10	11

FIG. 19.

EFFECT OF LENGTH ON COLLAPSING PRESSURE.— Abstract from Log of Tests Conducted by Prof. R. T. Stewart, 1902-04, on National Tube Co.'s Lap-welded Bessemer Steel Tubes in Lengths of 2½, 5, 10, 15, and 20 Feet, 8⅝" Outside Diameter, to Determine the Effect of Length of Tube on the Collapsing Pressure; to which is Added a Comparison with Values Read from the Curve, M, Representing the Average Results of these Tests. Made by E. E. S. under direction of R. T. S., 1905.

Tests Grouped and Arranged in Order of Length and Thickness of Tubes.

Number of Test	Actual Outside Diameter, Inches	Thickness of Wall, Inches		Unsupported Length of Tube Feet	Actual Plain End Weight, Lbs per Ft	Collapsing Pressure Pounds per Square Inch.				Designation of Tube, as Reported.
		Nominal	Computed from Wgt.			Observed	Corrected to Nom. Thk.	From Curve M	% Variation from M.	
52	8.664	0.271	0.258	19.649	23.13	1320	1450		+ 2	8¼" Casing, 24.38 lbs.
54	8.660	0.271	0.262	19.639	23.52	1485	1580		+11.5	" " " "
99	8.663	0.322	0.264	19.646	23.66	1375	1450		+ 2	8" Line Pipe, 28.18 lbs.
*79	8.664	0.281	0.266	17.986	23.84	1425	1480		+ 2.5	" " " 25.00 lbs.
50	8.660	0.271	0.271	19.646	24.28	1435	1435		+ 1	8⅝" Casing, 24.38 lbs.
53	8.660	0.271	0.272	19.638	24.32	1520	1510	1420	+ 6.5	" " " lb
76	8.663	0.281	0.272	19.634	24.38	1410	1400		- 1.5	8" Line Pipe, 25.00 lbs.
51	8.668	0.271	0.274	19.634	24.54	1430	1395		- 2	8⅝" Casing, 24.38 lbs.
75	8.640	0.281	0.274	19.649	24.44	1375	1345		- 5.5	8" Line Pipe, 25.00 lbs.
*77	8.660	0.281	0.274	19.501	24.47	1275	1245		-12.5	" " " " "
*78	8.660	0.281	0.280	18.396	25.08	1250	1145		- 20	" " " " "
	8.660		0.268	19.642	24.03	1419	1446		+ 1.2	20' Lengths, Corrected to Thk. .271 * Iron, not in averages.
118	8.645		0.294	2.158	26.22	2450	2780		+10.5	
122	8.645		0.298	2.160	26.60	2865	2655		+ 5.5	2½ Foot Lengths.
121	8.657	0.322	0.319	2.170	28.43	2290	2325	2515	- 7.5	
119	8.641		0.322	2.146	28.60	2490	2490		- 1	8" Line Pipe, 28.18 lbs.
120	8.640		0.323	2.150	28.64	2390	2380		- 5.5	
	8.646		0.311	2.157	27.70	2397	2526		+ 0.4	
115	8.658		0.297	4.645	26.48	1980	2270		- 1	
117	8.634		0.299	4.641	26.58	2325	2595		+13	5 Foot Lengths,
113	8.643	0.322	0.307	4.640	27.30	1795	1970	2290	-14	
114	8.673		0.307	4.635	27.43	2040	2220		- 3	8" Line Pipe, 28.18 lbs.
116	8.663		0.321	4.650	28.60	2225	2240		- 2	
	8.659		0.306	4.642	27.28	2073	2259		- 1.4	
110	8.626		0.306	9.640	27.20	2055	2235		- 4.5	
†112	8.672	0.322	0.314	9.645	28.04	1585	1675	2135	-21.5	10 Foot Lengths.
†108	8.652		0.325	9.640	28.70	1780	1750		+ 4	8" Line Pipe, 28.18 lbs.
	8.626		0.306	9.640	27.20	2055	2235		- 4.5	† O.H. Steel, not in averages.
123	8.665	0.281	0.297	14.641	26.49	1675	1940		- 4.5	8" Line Pipe, 25.00 lbs.
105	8.648	0.322	0.297	14.645	26.46	1520	1785		-12.5	" " " 28.18 lbs.
106	8.672	0.322	0.308	14.635	27.48	1685	1835	2035	- 9.5	" " " " "
103	8.641	0.322	0.320	14.640	28.38	2000	2025		- 0.5	" " " " "
107	7.666	0.322	0.324	14.630	28.89	2025	2005		- 1.5	" " " " "
	8.658		0.307	14.638	27.54	1781	1918		- 5.6	15 Foot Lengths.
101	8.647		0.294	19.641	26.27	1650	1935		- 1.5	
100	8.646		0.297	19.645	26.44	1710	1970		+ 0.5	
424	8.671		0.302	19.885	26.98	1575	1780		- 9.5	
423	8.649		0.303	19.792	27.00	1830	2025		+ 3	20 Foot Lengths.
421	8.666	0.322	0.303	19.760	27.09	1935	2135	1965	+ 8.5	
102	8.656		0.303	19.658	27.00	1960	2160		+10	8" Line Pipe, 28.18 lbs.
422	8.672		0.307	19.765	27.39	1635	1790		- 9	
†98	8.666		0.308	19.636	27.44	1735	1890		- 4	
420	8.658		0.310	19.875	27.64	1805	1930		- 1.5	
	8.658		0.302	19.753	26.98	1762	1966		+ 0.1	
1	2	3	4	5	6	7	8	9	10	11

Note— To obtain collapsing pressures for curve N, multiply corresponding tabular collapsing pressures from curve M by 0.944. Curve M is based upon Series One only, while curve N is based jointly upon both Series One and Series Two.

FIG. 20.

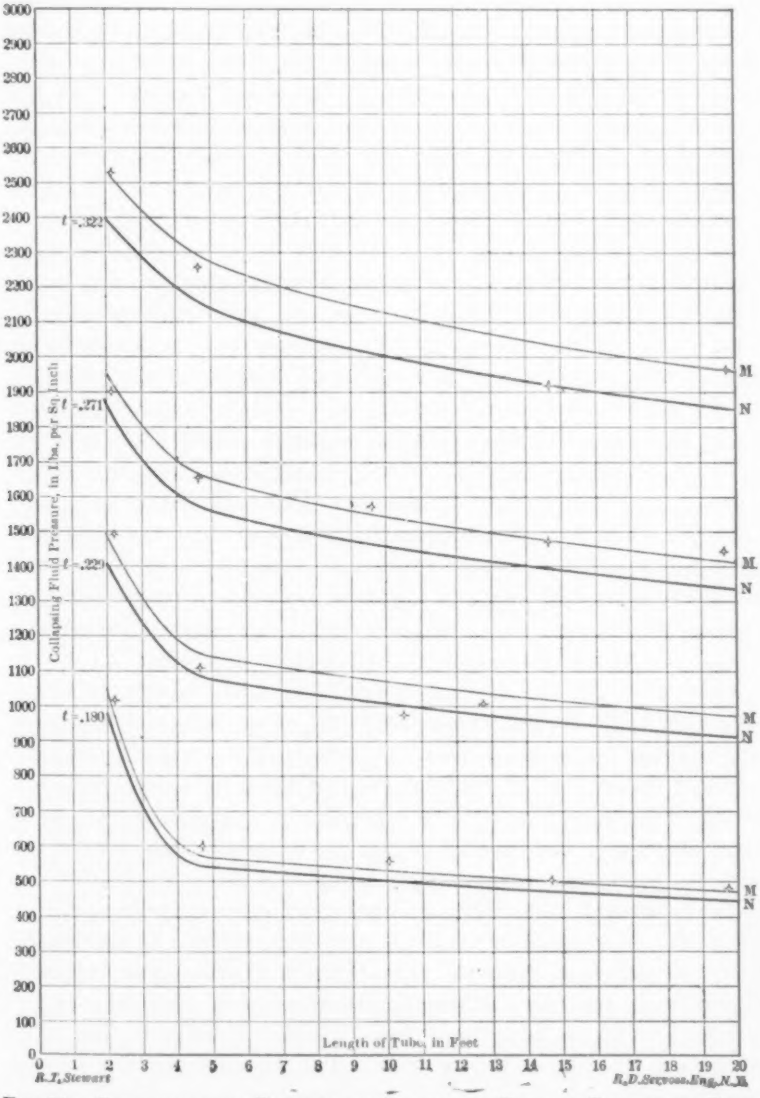


Fig. 21.—Chart showing Relation of Length of Tube to Collapsing Pressure for National Tube Co.'s Lap-welded Bessemer Steel Tubes. Based upon Tests on Lengths of 2½, 5, 10, 15, and 20 Feet, Conducted by Prof. R. T. Stewart, 1902–4.

The lines marked M show the relation of length to collapsing pressure, from tests on tubes 8⅝″ O.D., and the lines N, 5.6% below, this relation as based on all tests on outside diameters from 3 to 10 inches.

for each of the five lengths of tube tested, the actual average thickness of wall of each experimental tube shall fall in the group having the nearest nominal thickness of wall.

Collapsing Pressure Corrected to Nominal Thickness.—In the seventh column, or the first of those headed "Collapsing Pressure," Figs. 18-20, are entered the observed collapsing pressures. These pressures, of course, in each case correspond to the actual thickness of the tube. Since the actual thicknesses of different commercial tubes of the same nominal thickness of wall vary somewhat in practice, as is evident from running the eye down column 4 of this table, it became necessary, in order to get strictly comparable results, to obtain from the observed collapsing pressure of each experimental tube, whose thickness of wall did not equal exactly the nominal thickness, a collapsing pressure that would correspond to this nominal thickness. That is to say, the observed collapsing pressures corresponding to the respective thicknesses of wall tested were corrected, so that each would represent what the collapsing pressure would have been had the tube had the exact nominal thickness instead of that tabulated in column 4. These collapsing pressures, having been thus corrected to correspond to the nominal thickness of wall, are entered in column 8, or the second of those headed "Collapsing Pressure."

This correction was made graphically by first plotting, to a vertical scale representing collapsing pressure and a horizontal scale representing thickness of wall, the results of the tests for all the different thicknesses of wall for each of the five lengths of tube; second, drawing the mean line representing average results; and, third, drawing a line parallel to this mean line through the plotted value for each tube intersecting the ordinate corresponding to the thickness of that tube. The collapsing pressure corresponding to this point of intersection was then read and recorded in column 8.

The *Collapsing Pressure from Curve M* for each nominal thickness, in the five different lengths tested, are entered in column 9. These were read from a chart similar to Fig. 21, but drawn to a much larger scale. The variation of the values given in column 8 from the corresponding values in column 9, in per cent., are given in column 10. This column shows at a glance how the individual tests corrected to nominal thickness differ from the mean values as read from curve M. When it is considered that these were the ordinary commercial lap-welded wrought-tubes, selected at random, and subjected to an external

fluid pressure, it is surprising that there should be such a slight variation. It will be observed that the greatest individual variation does not exceed 22 per cent., while the greatest variation among the group averages is 8.2 per cent., the greater portion of these being less than 5 per cent. The average variation of the group averages is 0.2 per cent. Theoretically, of course, this should be zero, but the very small value of 0.2 per cent. obtained serves as a satisfactory check upon the accuracy of the mean values read from curve M.

Relation of Collapsing Pressure to Length of Tube.—This relation is clearly shown in Fig. 21 for $8\frac{1}{4}$-inch casing ($8\frac{5}{8}$ inches outside diameter) in the four nominal thicknesses of 0.180, 0.229, 0.271, 0.322 inch, and for lengths of from 2 to 20 feet between the regular screwed couplings.

On this chart the combined circles and crosses represent the different plotted averages contained in column 8, Figs. 18-20. By means of these plotted values the curves marked M were constructed. The spacing of these curves was adjusted so that when the values of the table, Figs. 18-20, were also plotted to collapsing pressure and thickness, for each of the five lengths tested, the resulting curves were smooth. By this method of cross-plotting the individual experiments on $8\frac{5}{8}$-inch outside diameter tubes, the four curves marked M were obtained. Each of these curves, then, is not based only upon the group averages belonging to it, but is also based upon the group averages belonging to the other three curves similarly marked.

The *curves marked N* were obtained by adjusting the curves M so as to harmonize with the most probable values for collapsing pressure as based upon Series Two. The difference, it will be observed, is small, the different curves N being only 5.6 per cent. below the corresponding curves M.

Discussion of Curves M and N.—An inspection of Fig. 21 discloses the fact that for this size of tube, especially for the thinner walls, there is a marked dropping off in collapsing pressure as the length of tube is increased up to about $4\frac{1}{2}$ feet, or until a length equal to about 6 diameters is reached. Beyond this point there appears to be no further material decrease in the collapsing pressure. For example, from curve M, for a thickness of 0.180 inch, the collapsing pressures for lengths of 2, $4\frac{1}{2}$ and 20 feet are respectively 1,050, 570, 480 pounds. That is to say, an increase in length from 2 to $4\frac{1}{2}$ feet diminishes the collapsing pressure by

480 pounds, while a further increase from $4\frac{1}{2}$ to 20 feet in length diminishes the collapsing pressure by only 90 pounds. As the thickness of wall is increased this disparity between the relative strength of long and short tubes becomes less prominent until for a thickness of 0.322 inch the difference is so small as to be of no practical importance. For example, assuming 20 feet to be the standard length of tube between end connections tending to hold it to a circular form, we find from curve N, for a thickness of 0.180 inch, that for lengths of 20, 15, 10, 5 and 2 feet, the respective collapsing pressures would be 450, 470, 500, 540, and 980 pounds, which correspond to increasees of 4.5, 11, 20, and 118 per cent. respectively; while for a thickness of 0.322 inch the values of the collapsing pressures for the same lengths are 1,850, 1,915, 2,000, 2,140 and 2,390 pounds, which correspond to increases of 3.5, 8, 16, and 29 per cent. respectively.

A study of these curves M and N, taken in connection with the photographs of the experimental tubes after being collapsed, and column 19 of the tabular statements of principal results of tests, will show conclusively that for lengths greater than about six diameters the strength of a tube when subjected to a fluid collapsing pressure is substantially constant. It must be remembered, while studying the photographs and column 19 of the tables referred to, that the inability to stop the hydraulic pressure pump instantly the experimental tube failed, together with the recoil of the hydraulic test cylinder, will ordinarily account for an extension of the length of the collapsed portion by probably two or more diameters, after failure had actually occurred.

Previously Published Formulae for the Collapsing Pressures of Tubes.

Preparatory to entering upon the experimental investigation of which this is a report, an extensive search was made through the technical literature where one would expect to find matters relating to the collapsing pressures of tubes and flues. As a result of this search a number of formulæ were collected and compared.

Since completing the present research comparisons were made of the results of the actual tests and the corresponding values calculated by the different published formulæ. These are shown in Figs. 22 to 33 inclusive. It will be observed that these comparisons have been made for plain tubes, $8\frac{5}{8}$ inches outside

diameter, in four commercial thicknesses, and for lengths of $2\frac{1}{2}$, 5, 10, 15, and 20 feet between end connnections tending to hold them to a circular form.

In the table and charts above referred to and in the discussion that here follows:

P = probable fluid collapsing pressure, in pounds per square inch, as based upon the present research, and for the conditions stated.

p = collapsing pressure as calculated by the different published formulæ for the same condition as for P.

d = outside diameter of tube in inches.

t = thickness of wall in inches.

l = length of tube in inches.

L = length of tube in feet.

P/p = The relation of the actual collapsing pressure, for any stated conditions, to that calculated by the different published formulæ for the same conditions.

Fairbairn's Formula.—From his own experiments Fairbairn established the following empirical formula:

$$p = 9{,}676{,}000\frac{t^{2.19}}{ld} = 806{,}300\frac{t^{2.19}}{Ld}.$$

He states that "the above is the general formula for the calculation of the strength of wrought-iron tubes subjected to external pressure, within the limits indicated by the experiments, that is, provided that the length is not less than 1.5 feet, and not greater than probably 10 feet." It would appear that this upper limit was arbitrarily fixed, since none of the tubes tested by him exceeded about five feet in length, and were held rigidly to a circular form at the ends.

Fig. 23 shows, plotted to scale, the values entered in columns 2 and 3 of Fig. 22. In this chart the vertical scale represents fluid collapsing pressure, in pounds per square inch, and the horizontal scale, length of tube, in feet, between end connections tending to hold it to a circular form.

The four curves shown in full lines are based upon the tests conducted, during the present research, on $8\frac{5}{8}$-inch outside diameter tubes in the four different thicknesses of wall shown and for the five lengths above stated. These lines are the same as curves N. (See page 767 and Fig. 21.)

The broken and dotted lines represent the results obtained by

Length of Tube in Feet.	Values from Curve N, Based on Tests.	Fairbairn's Formula, $p=9,676,000\frac{t^{2.19}}{ld}$		Fairbairn's Approx. Formula, $p=9,676,000\frac{t^2}{ld}$		Nystrom's Formula, $p=692,800\frac{t^2}{d\sqrt{l}}$		Unwin's Formula for Butt-joint, $p=9,614,000\frac{t^{2.21}}{l^{.9}d^{1.16}}$		Unwin's Formula for Lap-joint, $p=7,363,000\frac{t^{2.1}}{l^{.9}d^{1.16}}$		Grashof's Formula, $p=1,034,000\frac{t^{2.081}}{l^{.309}d^{.88}}$	
l	P	p	P/p	p	P/p	p	P/p	p	P/p	p	P/p	p	P/p
	$t=.180$	$t=.180$		$t=.180$		$t=.180$		$t=.180$		$t=.180$		$t=.180$	
2½	826	875	0.94	1212	0.68	475	1.74	836	0.99	773	1.07	631	1.31
5	535	437	1.22	606	0.88	336	1.59	448	1.19	414	1.29	426	1.26
10	502	219	2.29	303	1.66	238	2.11	240	2.09	222	2.26	288	1.74
15	472	*146	3.23	*262	2.39	194	2.43	†167	2.83	†154	3.06	230	2.05
20	449	*109	4.12	*151	2.97	168	2.67	†124	3.48	†119	3.77	195	2.30
	$t=.229$	$t=.229$		$t=.229$		$t=.229$		$t=.229$		$t=.229$		$t=.229$	
2½	1313	1482	0.89	1961	0.67	769	1.71	1423	0.92	1282	1.02	1041	1.26
5	1076	741	1.45	980	1.10	544	1.98	763	1.41	687	1.57	704	1.53
10	1010	370	2.73	490	2.06	385	2.62	409	2.47	368	2.74	476	2.12
15	952	*247	3.85	*327	2.91	314	3.03	†284	3.35	†256	3.72	379	2.51
20	915	*185	4.95	*245	3.73	272	3.36	†219	4.18	†197	4.64	322	2.84
	$t=.271$	$t=.271$		$t=.271$		$t=.271$		$t=.271$		$t=.271$		$t=.271$	
2½	1771	2143	0.83	2747	0.64	1877	1.64	2065	0.86	1826	0.97	1478	1.20
5	1561	1071	1.46	1373	1.14	761	2.05	1106	1.41	979	1.59	999	1.56
10	1460	536	2.72	687	2.13	539	2.71	593	2.46	525	2.78	676	2.16
15	1387	*357	3.89	*458	3.03	440	3.15	†412	3.37	†364	3.81	538	2.58
20	1337	*268	4.99	*343	3.90	380	3.52	†318	4.20	†281	4.76	457	2.93
	$t=.322$	$t=.322$		$t=.322$		$t=.322$		$t=.322$		$t=.322$		$t=.322$	
2½	2335	3127	0.75	3878	0.60	1521	1.54	3023	0.77	2623	0.89	2115	1.10
5	2140	1563	1.37	1939	1.10	1075	1.99	1620	1.32	1405	1.52	1430	1.50
10	2003	782	2.56	969	2.07	761	2.63	868	2.31	753	2.66	968	2.07
15	1915	*521	3.68	*646	2.96	621	3.08	†603	3.18	†523	3.66	770	2.49
20	1850	*391	4.73	*485	3.81	538	3.44	†465	3.98	†404	4.58	655	2.82

Length of Tube in Feet.	Wehage's Formula for Butt-joint, $p=494,000\frac{t}{d}\sqrt{\frac{t}{ld}}$		Wehage's Formula for Butt-joint, $p=253,600\frac{t}{d}\sqrt{\frac{t}{ld}}$		Clark's Formula, $p=t^2\left(\frac{50000}{d}-500\right)$		Loves Formula, $p=5,358,000\frac{t^2}{ld}+41,900\frac{t^2}{d^2}+1320\frac{t}{d}$		Belpaire's Formula, $p=3,427,000\frac{t^2}{ld}+56,890,000\frac{t^3}{ld^2}$		Eng. Board of Trade, for Working Pressure, $p=\frac{90,000t^2}{(l+1)d}$	
l	p	P/p	p	P/p	p	P/p	p	P/p	p	P/p	p	Factor of Safety
	$t=.180$		$t=.180$		$t=.180$		$t=.180$		$t=.180$		$t=.180$	
2½	2074	0.40	1073	0.77	172	4.80	856	0.96	573	1.44	97	8.52
5	1647	0.32	850	0.63	172	3.11	520	1.03	286	1.87	56	9.55
10	1306	0.38	675	0.74	172	2.92	353	1.42	143	3.51	31	16.19
15	1141	0.41	590	0.80	172	2.74	297	1.59	95	4.97	21	22.47
20	1037	0.43	537	0.84	172	2.61	269	1.67	72	6.24	16	28.06
	$t=.229$		$t=.229$		$t=.229$		$t=.229$		$t=.229$		$t=.229$	
2½	2860	0.46	1480	0.89	278	4.72	1376	0.95	993	1.32	156	8.42
5	2270	0.47	1175	0.92	278	3.87	833	1.29	496	2.17	91	11.82
10	1801	0.56	930	1.09	278	3.63	561	1.80	248	4.07	50	20.20
15	1574	0.60	823	1.16	278	3.42	471	2.02	165	5.77	34	28.00
20	1430	0.64	740	1.24	278	3.29	426	2.15	124	7.38	26	35.19
	$t=.271$		$t=.271$		$t=.271$		$t=.271$		$t=.271$		$t=.271$	
2½	3580	0.49	1855	0.95	389	4.55	1919	0.92	1467	1.21	219	8.09
5	2841	0.55	1470	1.06	389	4.01	1158	1.35	773	2.02	128	12.20
10	2255	0.65	1165	1.25	389	3.75	778	1.88	367	3.98	70	20.86
15	1969	0.70	1025	1.35	389	3.57	651	2.13	244	5.68	48	28.90
20	1789	0.75	925	1.45	389	3.44	588	2.27	183	7.31	36	37.14
	$t=.322$		$t=.322$		$t=.322$		$t=.322$		$t=.322$		$t=.322$	
2½	4564	0.52	2335	1.00	549	4.25	2700	0.86	2202	1.06	309	7.56
5	3575	0.60	1852	1.16	549	3.90	1626	1.32	1101	1.94	180	11.89
10	2837	0.71	1465	1.37	549	3.65	1090	1.84	550	3.64	98	20.44
15	2479	0.77	1283	1.49	549	3.49	911	2.10	367	5.22	68	28.16
20	2251	0.82	1165	1.59	549	3.37	821	2.25	275	6.73	51	36.27

*Beyond Fairbairn's limit of length. †Beyond Unwin's limit of length.

FIG. 22.—COMPARISON OF VALUES FOR THE COLLAPSING PRESSURES OF 8⅝" O. D. TUBES, CALCULATED BY THE DIFFERENT PUBLISHED FORMULAE, WITH CORRESPONDING VALUES BASED UPON TESTS BY PROF. R. T. STEWART.

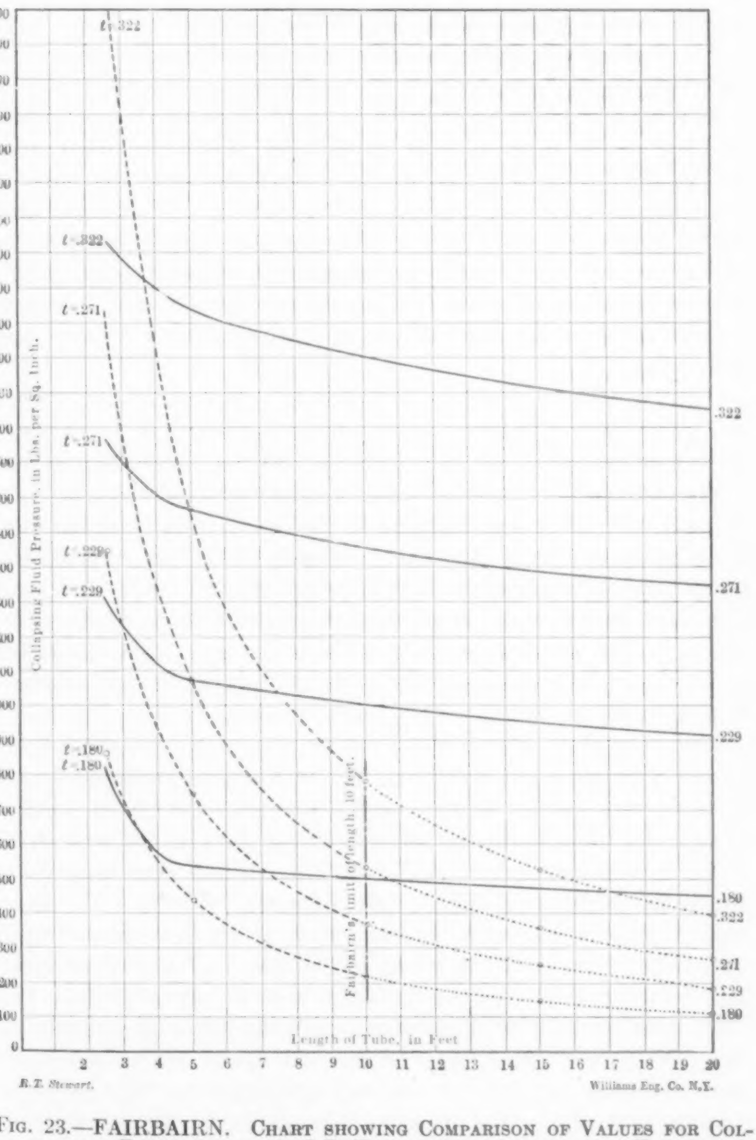


FIG. 23.—FAIRBAIRN. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF 8⅝″ O.D. TUBES, OBTAINED BY USE OF FAIRBAIRN'S FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Fairbairn's Formula, full lines those based on Prof. Stewart's experiments.

Fairbairn states that his formula is applicable to lengths from 1½ to 10 feet.

plotting to the same scales the corresponding values for collapsing pressure as calculated by Fairbairn's formula.

For both sets of curves the thickness of wall in decimals of an inch is written at both ends of the curve representing the tube.

It should be observed that the portions of the curves consisting of long dashes represent the application of Fairbairn's fomula within the limits of length assigned by him, while the portions consisting of short dashes are beyond this limit. These latter have been drawn for the purpose of showing how utterly inapplicable is this formula to modern commercial tubes of comparatively long lengths between joints or end supports tending to hold them to a circular form, such as well casing, boiler tubes and long, plain boiler flues.

Applicability of Fairbairn's Formula.—It would appear from this chart that Fairbairn's formula is applicable only to tubes having a comparatively thin wall and, at the same time, a relatively short length between joints tending to hold the tube to a circular form. That this should be the case is quite natural, since the experiments furnishing the data upon which this formula was based involved these conditions of relatively thin walls and short lengths. This is quite apparent from an inspection of the chart which, it will be observed, is for tubes having outside diameters of $8\frac{5}{8}$ inches in lengths of from $2\frac{1}{2}$ to 20 feet, and for thicknesses of wall of 0.180, 0.229, 0.271 and 0.322 inch.

The only place on the chart where values calculated by Fairbairn's formula agree substantially, over any appreciable length of tube, with those obtained by the present research on modern commercial tubes is for the tubes having a thickness of wall equal 0.180 inch, which were the thinnest tested, and then only for a length up to about 4.5 feet, or six diameters of tube. The chart shows that at Fairbairn's limit of length of 10 feet a tube having an outside diameter of $8\frac{5}{8}$ inches and a thickness of wall of 0.180 inch would probably collapse at 500 pounds per square inch. For these same conditions Fairbairn's formula gives a pressure of 220 pounds, or 44 per cent. of this value, the actual collapsing pressure being thus 2.3 times that calculated by this formula. Again, for a plain tube of twice this length, or 20 feet between end connections tending to hold it to a circular form, and for the same diameter and thickness of wall, we get 450 pounds for the former and 110 pounds for the latter. For this case it appears that the value calculated by use of Fairbairn's formula is only

24 per cent. of that obtained by experiment, the latter for this case being 4.1 times the former. It is thus seen that when applied to an $8\frac{5}{8}$-inch tube, 0.180 inch thick, and 20 feet long between joints, Fairbairn's formula gives a result for the collapsing pressure that is a trifle over 300 per cent. in error.

It will be observed that Fairbairn's formula is even less applicable to a tube having a thicker wall. For example, Fig. 23 shows that for the same diameter of tube as before, but having a thickness of wall equal 0.271 instead of 0.180 inch, the true values for probable collapsing pressures at Fairbairn's limit and for lengths of 20 feet are, respectively, 2.7 and 5.0 times that obtained by use of his formula. In other words, this formula for these conditions gives results that are apparently in error by respectively 170 and 400 per cent.

For $8\frac{5}{8}$-inch commercial tubes having thicker walls than 0.180 inch it will be observed that Fairbairn's formula gives the correct collapsing pressure for but one length, namely, that at which the broken line cuts the corresponding full line. The chart shows clearly that Fairbairn's formula does not apply to modern lap-welded tubes having either relatively thick walls or long lengths between joints or end connections tending to hold them to a circular form.

Material of Fairbairn's Tubes.—No attempt has been made, in this report, to modify the constant of Fairbairn's formula so as to adapt it to steel tubes. This was because of two reasons: First, no determination appears to have been made for the physical properties of the wrought iron constituting Fairbairn's experimental tubes, at least an examination of the records of his research has not disclosed any information on this point. Second, even had we a record of the physical properties of the material of his tubes, there appears to be no simple relation between the collapsing pressure of a tube and the physical properties of the material of which it is formed. For a discussion of this, see page 800.

However, since Fairbairn's experimental tubes were made from rolled plates, No. 19 B.W.G., or 0.042 inch thick, of presumably high-grade iron, the flat plates being rolled cold into tubular form, then riveted and brazed, it would appear that, for these conditions, the material of these tubes would not differ greatly in physical properties from those of the very soft steel used in modern commercial lap-welded tubes.

Fairbairn's Approximate Formula.—This formula was obtained

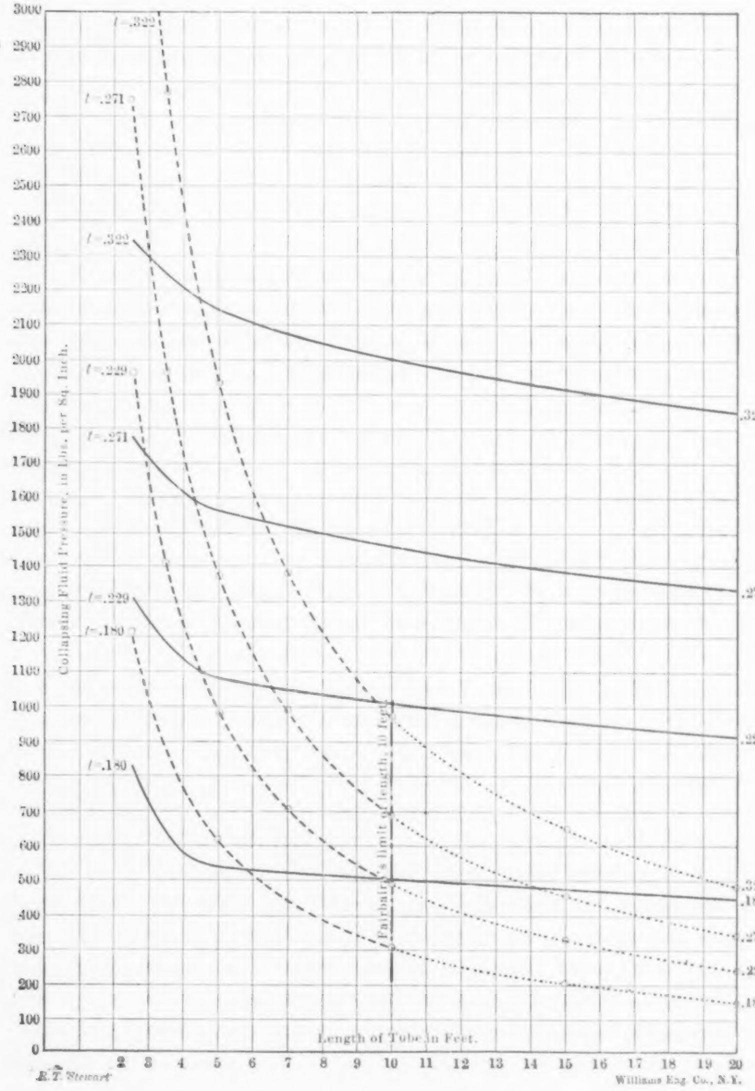


FIG. 24.—FAIRBAIRN. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF $8\frac{5}{8}''$ O.D. TUBES, OBTAINED BY USE OF FAIRBAIRN'S APPROXIMATE FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Fairbairn's Approximate Formula, full lines those based on Prof. Stewart's experiments.

Fairbairn states that his formula is applicable to lengths from $1\frac{1}{2}$ to 10 feet.

from the preceding more exact one by changing the factor $t^{2.19}$ to t^2, thus giving rise to a formula that could be more readily handled in the making of calculations. The formula thus modified is

$$p = 9{,}676{,}000 \frac{t^2}{ld} = 806{,}300 \frac{t^2}{Ld}.$$

The precise manner in which this change affects the results can best be had by making a comparison of Fig. 24 with Fig. 23. Also by comparing columns 5 and 6 with 3 and 4 of Fig. 22. Such a comparison will show that, while not representing the conditions of Fairbairn's experiments, namely, relatively thin walls and short lengths, as well as the more exact formula, it is also quite inapplicable to modern wrought tubes in relatively long lengths.

Grashof's Formula.—Grashof selected from Fairbairn's experiments 17 of those having walls 0.042-inch thick and 4 having walls $\frac{1}{4}$ to $\frac{1}{4}$ inch thick and from these 21 experiments he deduced the following formula:

$$p = 24{,}480{,}000 \frac{t^{2.315}}{ld^{1.278}} = 2{,}040{,}000 \frac{t^{2.315}}{Ld^{1.278}}. \quad . \quad . \quad .(A)$$

As this formula was found to represent the results of Fairbairn's experiments on the tubes having walls 0.042-inch thick better than those having walls $\frac{1}{8}$ to $\frac{1}{4}$ inch thick, he derived the following formula for the latter, namely:

$$p = 1{,}033{,}600 \frac{t^{2.081}}{l^{0.564}d^{0.889}} = 86{,}130 \frac{t^{2.081}}{L^{0.564}d^{0.889}}. \quad . \quad . \quad .(B)$$

This formula has been applied to tubes having an outside diameter equal $8\frac{5}{8}$ inches, in lengths from $2\frac{1}{2}$ to 20 feet and for four commercial thicknesses of wall from 0.180 to 0.322 inch. The precise manner in which these calculated values differ from the true probable collapsing pressures is clearly shown in Fig. 25 and in column 14 of the table, Fig. 22.

Nystrom's Formula.—Nystrom also used Fairbairn's experiments for the deduction of his formula for the collapsing strength of flues.

$$p = \frac{4Tt^2}{d\sqrt{L}}.$$

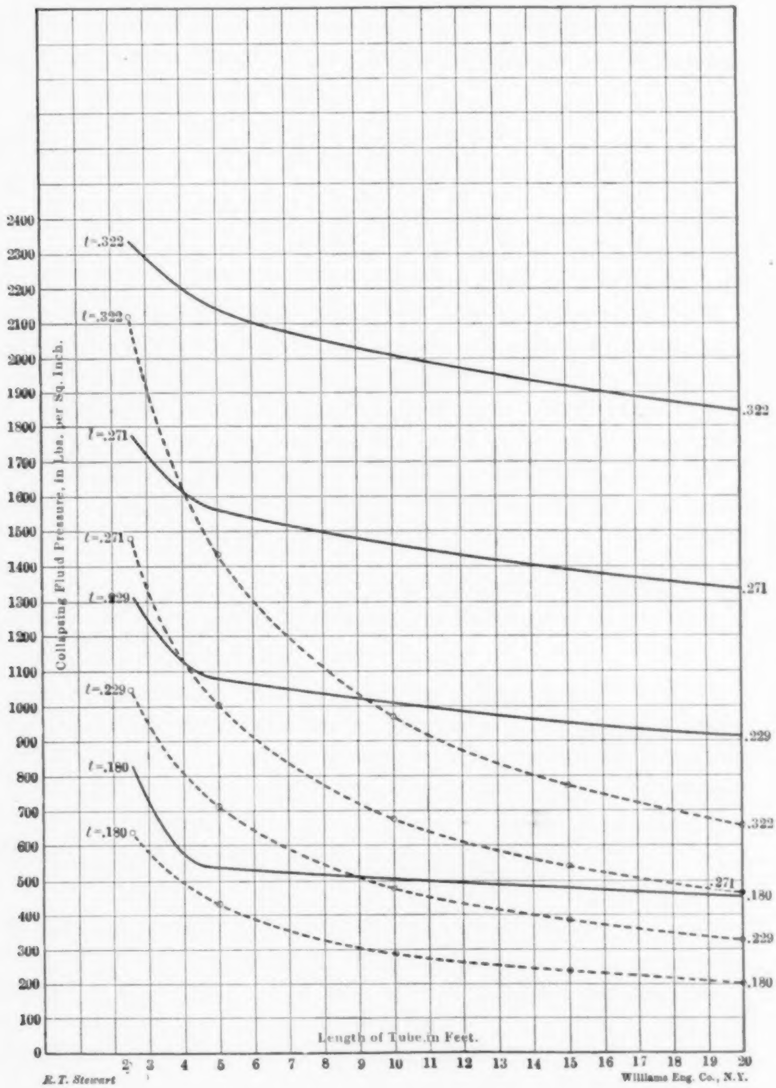


FIG. 25.—GRASHOF. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF $8\frac{5}{8}''$ O.D. TUBES, OBTAINED BY USE OF GRASHOF'S FORMULA B, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Grashof's Formula, full lines those based on Prof. Stewart's experiments.

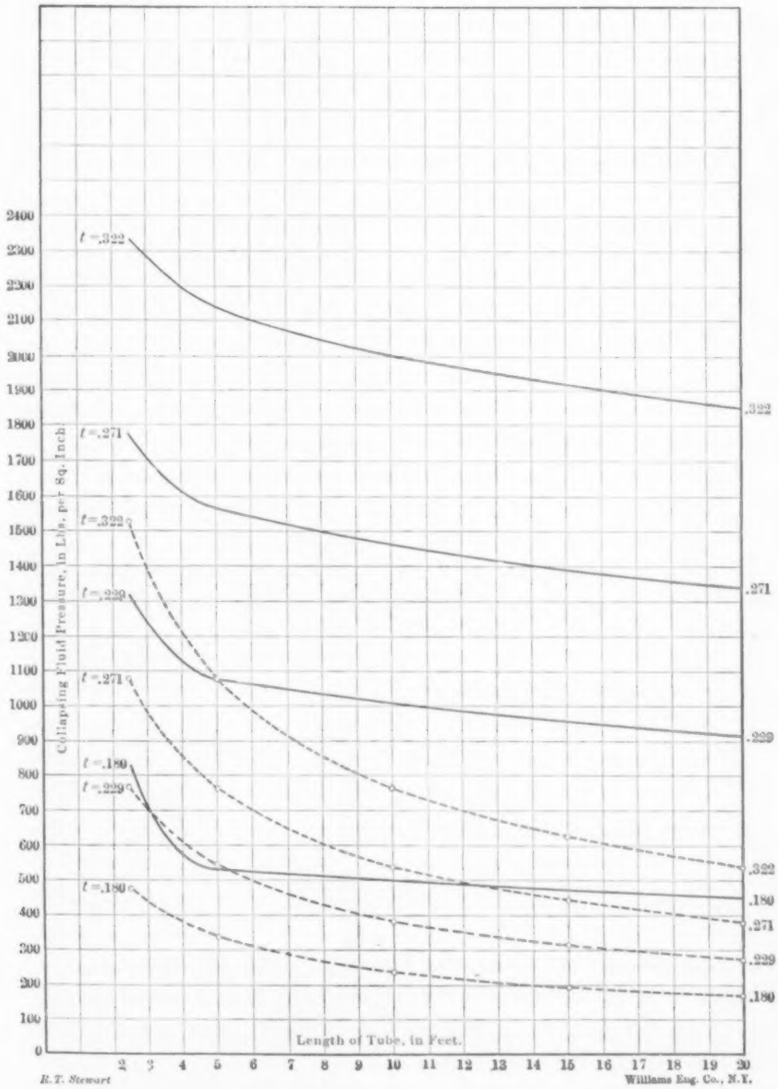


FIG. 26.—NYSTROM. CHART SHOWING COMPARISON OF VALUE FOR COLLAPSING PRESSURES OF 8⅝″ O.D. TUBES, OBTAINED BY USE OF NYSTROM'S FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Nystrom's Formula, full lines those based on Prof. Stewart's experiments.

Where T is the tensile strength of the material. Substituting 50,000 for T and $\frac{l}{12}$ for L in this formula gives

$$p = 692,800 \frac{t^2}{d\sqrt{l}}.$$

Nystrom considered 4 a sufficient factor of safety for use with his formula.

The customary value of 50,000 for T has been used in this formula for two reasons: (1) In order that the results obtained might be comparable with the other heretofore published formulæ, all of which are presumably for wrought iron; and (2) because, since the collapsing strength of a tube, in the light of the present research, appears to be quite independent of the tensile strength of the material constituting it, it was thought best not to attempt any modification of the formula. (See page 800.)

A comparison of pressures obtained from Nystrom's fomula and the probable collapsing pressures of modern lap-welded tubes is shown in Fig. 26 and column 8 of Fig. 22.

Unwin's Formulæ.—Unwin, who had been associated with Fairbairn when he made his collapsing tests on tubes, has derived the following formulæ for thick-walled tubes, namely, walls $\frac{1}{8}$ to $\frac{1}{4}$ inch thick.

For tubes with a longitudinal butt-joint:

$$p = 9,614,000 \frac{t^{2.21}}{l^{0.9}\, d^{1.16}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

For tubes with a longitudinal lap-joint:

$$p = 7,363,000 \frac{t^{2.1}}{l^{0.9}\, d^{1.16}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

Unwin states that when the length of tube between end connections or transverse joints tending to hold it to a circular form is at least 10 or 12 diameters, the strength does not decrease with further increase of length.

A comparison of values from these formulæ with the probable collapsing pressures of modern tubes is given in Figs. 27 and 28, and columns 10 and 12 of Fig. 22.

Clark's Formulæ.—For the derivation of his formula A, Clark selected, from the reports of the Manchester Steam-Users Association, the dimensions of six boiler flues which collapsed while in

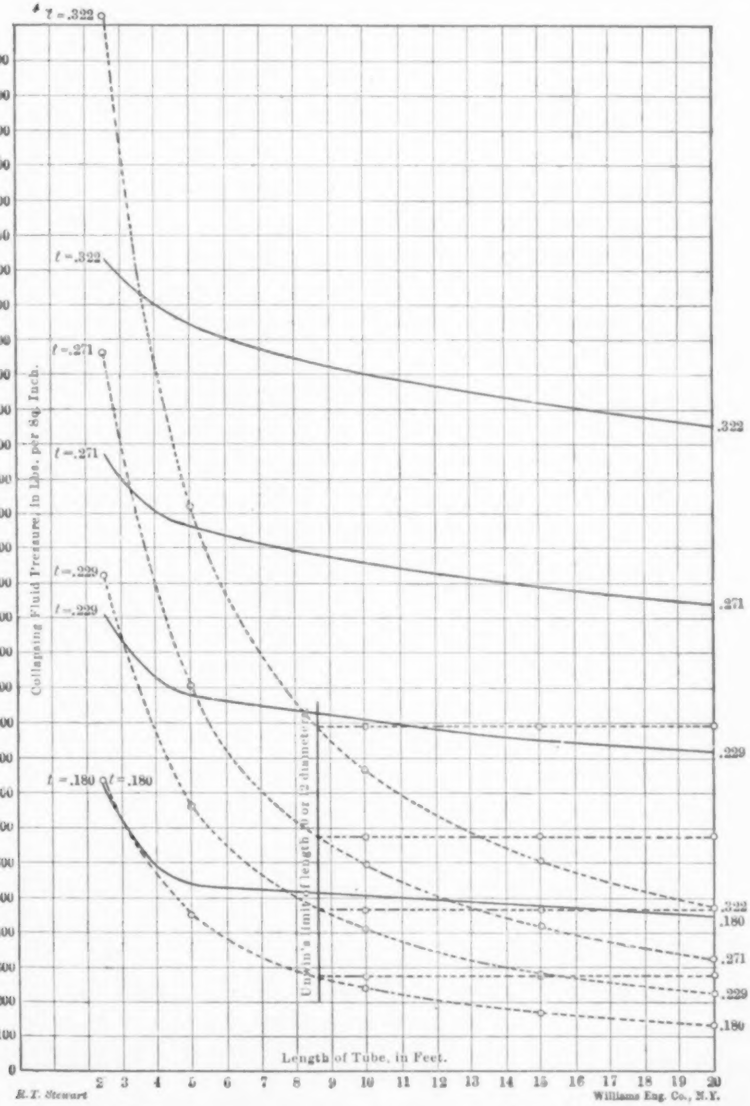


FIG. 27.—UNWIN. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF 8⅝″ O.D. TUBES, OBTAINED BY USE OF UNWIN'S FORMULA A, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Unwin's Formula "for tubes with a longitudinal butt-joint," full lines those based on Prof. Stewart's experiments.

Unwin states that when the length is at least 10 or 12 diameters the strength does not decrease with further increase of length.

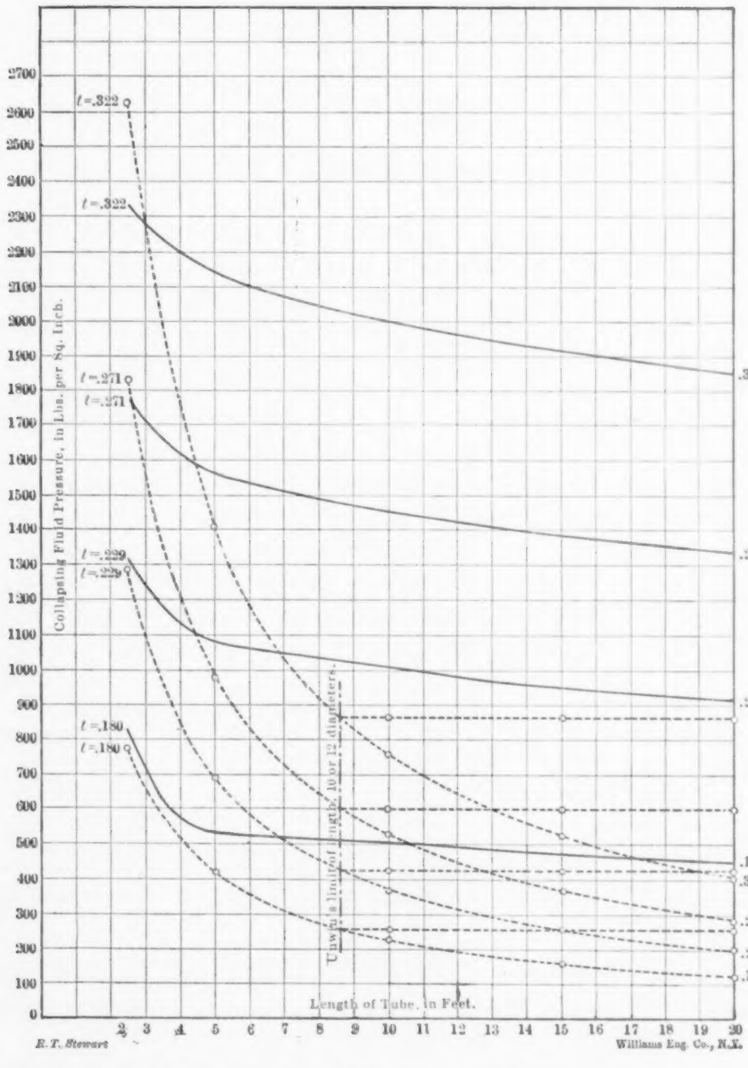


FIG. 28.—UNWIN. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF 8⅝" O.D. TUBES, OBTAINED BY USE OF UNWIN'S FORMULA B, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Unwin's Formula "for tubes with a longitudinal lap-joint," full lines those based on Prof. Stewart's experiments.

Unwin states that when the length is at least 10 or 12 diameters the strength does not decrease with further increase of length.

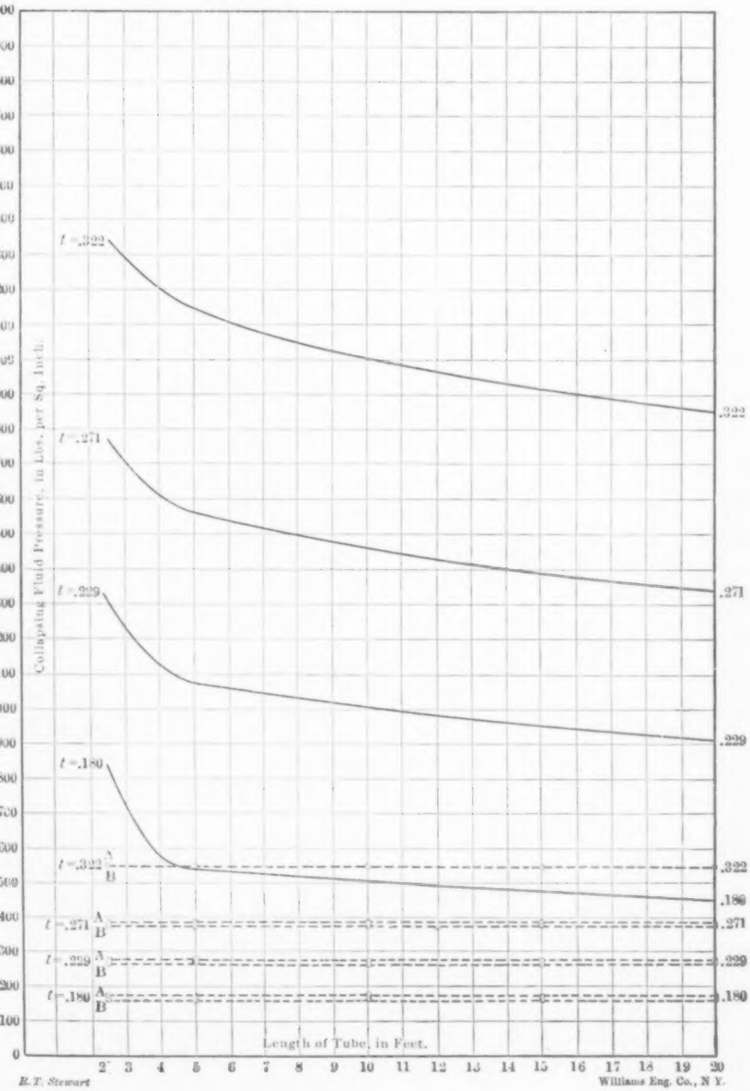


FIG. 29.—CLARK. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF $8\frac{5}{8}''$ O.D. TUBES, OBTAINED BY USE OF CLARK'S FORMULÆ, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Clark's Formulæ, A & B, full lines those based on Prof. Stewart's experiments.

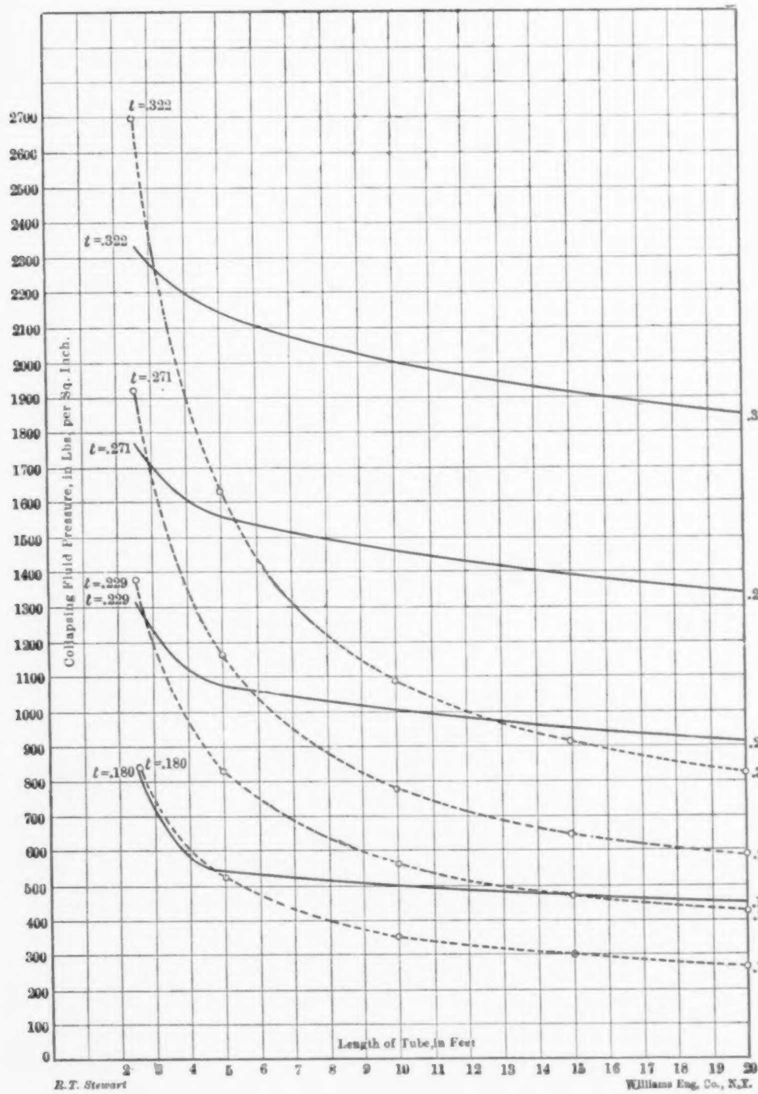


FIG. 30.—LOVE. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF $8\frac{5}{8}''$ O.D. TUBES, OBTAINED BY USE OF LOVE'S FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Love's Formula, full lines those based on Prof. Stewart's experiments.

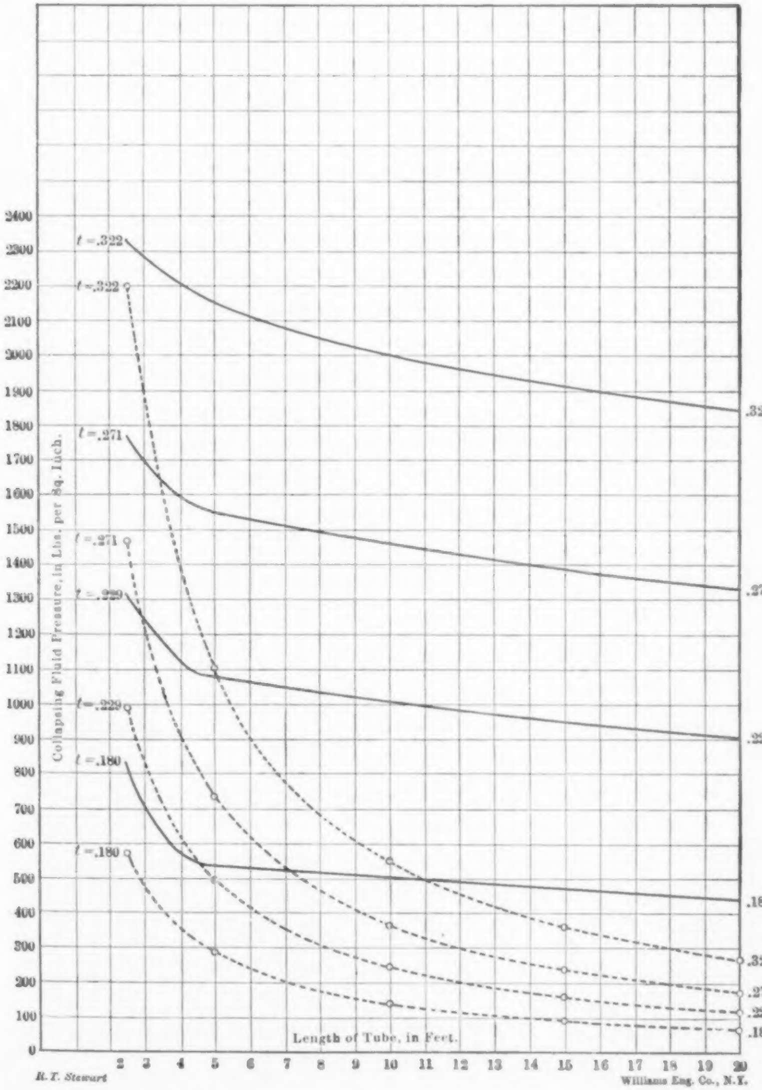


FIG. 31.—BELPAIRE. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURE OF $8\frac{5}{8}''$ O.D. TUBES OBTAINED BY USE OF BELPAIRE'S FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Belpaire's Formula, full lines those based on Prof. Stewart's experiments.

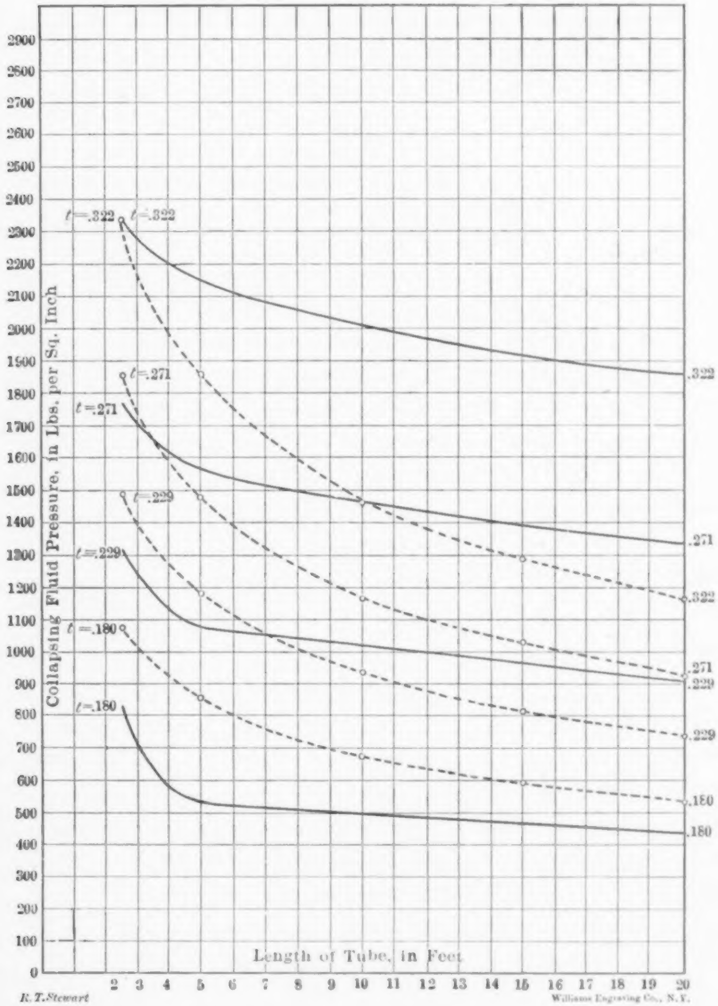


Fig. 32.—WEHAGE. Chart Showing Comparison of Values for Collapsing Pressures of 8⅝″ O.D. Tubes, obtained by Use of Wehage's Formula, with Values Based on Tests on National Tube Co.'s Bessemer Steel Lap-welded Tubes.

Broken lines show values obtained by use of Wehage's Formula for welded or butt-joints, from Dingler's Journal, Vol. 242, 1881, page 236, and the 4th German ed. Reuleaux's Const., page 1084. The full lines show values based on P of. Stewart's experiments.

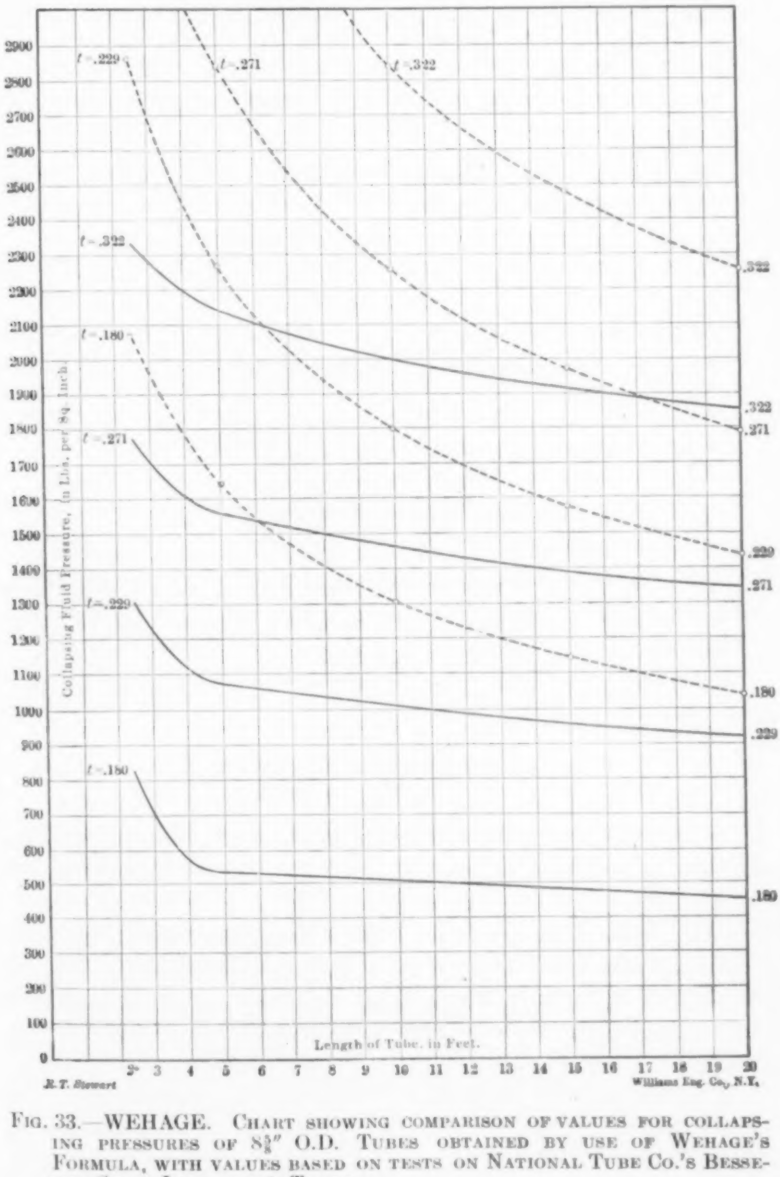


FIG. 33.—WEHAGE. CHART SHOWING COMPARISON OF VALUES FOR COLLAPSING PRESSURES OF $8\frac{5}{8}''$ O.D. TUBES OBTAINED BY USE OF WEHAGE'S FORMULA, WITH VALUES BASED ON TESTS ON NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES.

Broken lines show values obtained by use of Wehage's Formula for welded or butt-joints, from Releaux's Constructor translated by Suplee, 1893, page 269. The full lines show values based on Prof. Stewart's experiments.

actual use and in boilers under known pressures. The resulting formula was

$$p = t^2 \left(\frac{50,000}{d} - 500 \right) \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A)}$$

For the collapsing pressure of plain riveted boiler flues, Clark gives the following formula:

$$p = \frac{200,000\, t^2}{d^{1.75}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(B)}$$

For a comparison of values from these formulæ with the probable collapsing pressures of lap-welded tubes, see Fig. 29 and column 20 of Fig. 22.

Love's Formula.—M. Love's formula, which was also deduced from Fairbairn's experiments, is as follows:

$$p = 5,358,000 \frac{t^2}{ld} + 41,900 \frac{t^2}{d} + 1,320 \frac{t}{d}.$$

For a comparison of values obtained by its use with the collapsing pressures of modern tubes, see Fig. 30 and column 22 of Fig. 22.

Belpaire's Formula.—From Fairbairn's experiments Belpaire has deduced the following formula:

$$p = 3,427,000 \frac{t^2}{ld} + 56,890,000 \frac{t^3}{ld^2}.$$

For a comparison of values from this formula with the probable collapsing pressures of modern tubes, see Fig. 31 and column 24 of Fig. 22.

Dr. Wehage deduced a formula for flues with butt-joints which he states is also applicable to lap-welded tubes. This formula was apparently based upon Fairbairn's three experiments on tubes thicker than $\frac{1}{8}$ inch together with three isolated tests on boiler flues and two failures of flues while in service.

In metric units, as published in Dingler's Journal, vol. 242, 1881, page 236, and in the 4th German ed. of Reuleaux's Constructor, this formula is—

$$a^1 = 120,000 \frac{\delta}{D} \sqrt[3]{\frac{\delta}{lD}}$$

where the collapsing pressure, a^1, is expressed in kilograms per sq.

centimeter; while the diameter, D, the thickness, δ, and length l, are in millimeters.

Reduced to the same British units in which the other formulæ are expressed, this formula becomes

$$p = 253{,}600 \frac{t}{d} \sqrt[3]{\frac{t}{Ld}}.$$

A comparison of values obtained by use of this formula with the results of the present research is shown in Fig. 32 and in column 18 of Fig. 22.

Suplee's translation of Reuleaux's Constructor, 1893, page 269, states this formula in British units to be

$$p = 490{,}000 \frac{t}{d} \sqrt[3]{\frac{t}{Ld}}.$$

For a comparison of results by this formula with the results of the present research see column 16 of Fig. 22, and also Fig. 33.

English Board of Trade's Formula.—The following formula from the English Board of Trade is obtained from Fairbairn's approximate formula by using a factor of safety of about 9, and substituting L + 1 for L, namely:

$$p = \frac{90{,}000\, t^2}{(L+1)d}.$$

In column 26 of Fig. 22 will be found a comparison of values from this formula with the collapsing pressures of lap-welded tubes. It will be seen from an inspection of column 26 that the actual factor of safety, resulting from the application of this formula for the assumed conditions, varies approximately from 8 to 36.

Collapsing Tests, Series Two, on Twenty-foot Lengths, Showing the Influence of Diameter and Thickness of Wall on the Collapsing Pressure.

The purpose of Series One was to determine the precise nature of the influence of length of tube upon the collapsing pressure. By length of tube is here meant the distance between couplings, or other end connections, of a single length of tube, tending to hold

it to a circular form. The experimental determinations constituting Series One, as recorded elsewhere in this paper, see page 753 and Fig. 21, show conclusively that for commercial wrought tubes $8\frac{5}{8}$ inches outside diameter, there is no practical difference in the collapsing pressure for lengths greater than six diameters up to twenty feet. As soon as this point had been fully established experimentally it was decided to make all succeeding tests on tubes in lengths of 20 feet. This was accordingly done, the results of these tests being grouped as Series Two.

The *apparatus used,* and the manner of making the tests of Series Two were, in every essential respect, precisely the same as for Series One. A complete detailed statement of these will be found in that portion of this paper that deals with Series One.

The *tabular statement of principal results* of Series Two will be found in Figs. 34 to 42. This tabular statement is presented in exactly the same form as that of Series One. For the precise meaning of the different entries in this table, see the explanation of the entries of the corresponding table of Series One.

Derivation of Formulæ for the Probable Collapsing Pressure of Lap-welded Bessemer Steel Tubes for Lengths of Twenty Feet.

Re-grouping of Tests.—The first step taken toward the derivation of formulæ for the strength of wrought tubes subjected to external fluid pressure, as based upon the results of the present research, was the re-grouping of the tests as shown in Figs. 43-46.

This table contains an abstract from the Log of the results of all tests on tubes in lengths of 20 feet, excepting those that were intentionally dinged or put out of round, before being tested, for the purpose of obtaining data on the results of such defects.

It will be observed that the tests in this table are grouped according to the outside diameter of tube and, for each diameter, are arranged in the order of thickness of wall.

Plotting Group Averages to $\frac{t}{d}$.—It is apparent that for tubes subjected to external fluid pressure there are three principal variables involved, namely, the outside diameter, d, the thickness of wall, t, and the fluid-collapsing pressure, P. It is also apparent that each of these variables is a function of the other two, that is to say, depends jointly upon each of them for its value.

.SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

* These tests were made on speci…
and are therefore too high for Yield Point and …
The following are average values for 30 undis…
Tensile Strength, lbs p…
Yield Point, lbs per sq…
Elongation, % in 8 inche…
Reduction of Area, %…

Test Number	Outside Diameter Inches: Nominal	Outside Diameter: Average	Outside Diameter At Place of Collapse: Greatest	Outside Diameter At Place of Collapse: Least	Thickness of Wall Inches: Nominal	Thickness of Wall: Average	Thickness of Wall At Place of Collapse: Greatest	Thickness of Wall At Place of Collapse: Least	Length of Tube Feet: As Reported	Length of Tube: Actual	Length of Tube: Unsupported	Weight of Tube Lbs per Foot: Nominal	Weight of Tube: Actual	Collapsing Pressure: Pounds per Sq. Inch	Collapsing Pressure: Gage Used	Collapsing Pressure: Rate of Increase Lbs. per Sec.	Collapsed Length: In Feet	Collapsed Length: In Dia's.
200		6.018	6.030	5.950		0.123	0.135	0.106		20' 0¼"	19' 11⅛"		7.77	450		2	3' 0"	6
201		6.026	6.040	5.990		0.127	0.175	0.096		20' 0⅜"	19' 10⅝"		8.08	500		2	2' 9"	5.5
202	6.000	6.015	6.030	5.980	0.134	0.130	0.135	0.103	20' 0"	20' 0¼"	19' 10⅝"	8.26	8.19	575	B	2	3' 6"	7
203		6.017	6.020	5.960		0.130	0.137	0.110		20' 0½"	19' 10⅜"		8.19	540		2	3' 0"	6
204		6.010	6.020	5.940		0.131	0.147	0.116		20' 0⅜"	19' 10⅜"		8.24	530		1	3' 6"	7
Average		6.017	6.028	5.964		0.128	0.146	0.106		20' 0⅜"	19' 10⅝"		8.08	519		1.8	3' 2"	6.3
205		6.023	6.020	5.960		0.131	0.148	0.109		20' 0⅜"	19' 10⅝"		8.22	530		2	3' 2"	6.3
206		6.021	6.000	5.960		0.129	0.133	0.106		20' 0⅜"	19' 10⅜"		8.12	480		1	3' 6"	7
207	6.000	6.013	5.990	5.940	0.134	0.134	0.178	0.111	20' 0"	20' 0¼"	19' 10½"	8.26	8.45	640	B	2	4' 0"	8
208		6.013	6.010	5.920		0.135	0.136	0.110		20' 0"	19' 10⅜"		8.45	510		2	2' 9"	5.5
209		6.016	5.990	5.890		0.128	0.142	0.110		20' 0¼"	19' 10⅜"		8.07	485		1	3' 3"	6.5
Average		6.017	6.002	5.934		0.131	0.147	0.109		20' 0½"	19' 10⅜"		8.26	529		1.6	3' 4"	6.7
210		6.007	6.020	5.970		0.167	0.198	0.147		20' 0¼"	19' 10⅞"		10.44	1025	B	3	4' 1"	8.2
211		6.024	6.070	5.970		0.169	0.218	0.161		20' 0⅜"	19' 10⅞"		10.59	930	B	5	3' 6"	7
212	6.000	6.011	6.030	5.980	0.156	0.166	0.174	0.150	20' 0"	20' 0⅜"	19' 11⅛"	10.46	10.35	950	B	1	4' 2"	8.3
213		6.037	6.050	5.990		0.164	0.178	0.115		20' 0½"	19' 11⅛"		10.30	850	B	2	4' 0"	8
214		6.031	6.030	5.980		0.171	0.177	0.146		20' 0⅜"	19' 11⅛"		10.69	1090	C	2	5' 0"	10
Average		6.022	6.040	5.978		0.167	0.189	0.144		20' 0⅜"	19' 11⅛"		10.47	969		2.6	4' 2"	8.3
215		6.033	6.010	5.960		0.168	0.194	0.126		20' 0⅜"	19' 11"		10.55	1010		2	3' 6"	7
216		6.028	6.050	5.990		0.155	0.222	0.127		20' 0⅜"	19' 10⅞"		9.75	760		2	4' 0"	8
217	6.000	6.038	6.070	5.990	0.156	0.175	0.187	0.143	20' 0"	20' 0¼"	19' 11⅛"	10.46	10.92	1010	B	3	4' 0"	8
218		6.008	6.030	5.950		0.168	0.177	0.130		20' 0¼"	19' 11⅛"		10.50	860		2	4' 0"	8
219		6.021	6.040	5.980		0.166	0.199	0.131		20' 0⅜"	19' 10⅞"		10.37	780		2	4' 0"	8
Average		6.026	6.040	5.974		0.166	0.196	0.131		20' 0⅜"	19' 11"		10.42	924		2.2	3' 11"	7.8
221		6.034	6.080	5.990		0.166	0.167	0.114		20' 0⅜"	19' 10½"		10.40	760	E	2	3' 6"	7
222	6.000	6.037	6.090	5.900	0.156	0.156	0.173	0.130	20' 0"	20' 0¼"	19' 10½"	10.46	9.79	880	B	2	4' 0"	8
224		6.026	6.040	5.980		0.166	0.167	0.149		20' 0¼"	19' 10½"		10.40	1110	C	2	5' 0"	10
Average		6.032	6.070	5.957		0.163	0.170	0.131		20' 0⅜"	19' 10½"		10.20	917		2	4' 2"	8.3
223		6.026	6.040	6.000		0.169	0.179	0.152		20' 0⅜"	19' 9"		10.54	1070	C	5	3' 0"	6
225		6.038	6.050	5.970		0.166	0.190	0.117		20' 0⅜"	19' 8⅞"		10.39	715	B	3	3' 0"	6
226	6.000	6.028	6.100	5.990	0.156	0.178	0.168	0.137	20' 0"	20' 0¼"	19' 8⅞"	10.46	11.14	1425	C	5	2' 6"	5
238		6.037	6.060	5.990		0.167	0.180	0.110		20' 0"	19' 8⅞"		10.45	775	B	3	3' 6"	7
291		6.034	6.070	5.970		0.168	0.180	0.133		20' 0"	19' 8⅞"		10.50	1050	C	5	2' 6"	5
Average		6.033	6.064	5.984		0.170	0.179	0.130		20' 0⅜"	19' 8⅞"		10.60	1007		4.2	2' 11"	5.8
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 34.—TABULAR STATEMENT OF PRIN…

on specimens cut from distorted portions of the tube,
nt and too low for Elongation and Reduction of Area
30 undistorted specimens from groups 200 and 300.
h, lbs per sq in ---- 57,890.
per sq in. -------- 36,860
8 inches ---------- 21.18
ea, % ------------ 57.34

A - See General Remark Sheet Nº 33.

a & b - See types of Test Heads on General Remark Sheet Nº 33.

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R. T. STEWART, 1902-4   F.R.K., 1905.

…apsed Portion: …h Dia's.	Distance from End	Angular Distance from Weld	Physical Properties: Tensile Strength Lbs per Sq In	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Chemical Analysis %: Silicon	Sulphur	Phos.	Mang	Carbon	Oxide	Material	Remarks	Commercial Designation of Tube as Reported
6	2' 5½"	180°	*58 350	*37 470	*19 29	*54.80	—	067	114	.39	.08	—		b	
5.5	5' 6"	+150°	*56 220	*37 630	*17.25	*52 40	—	.060	110	.50	.08	—		A, b	
7	2' 4"	−150°	*58 330	*37 630	*17.71	*57.80	—	.066	.105	.44	.07	—	Bessemer Steel	b	6" Converse Joint 8.26 Lbs
6	2' 0"	−150°	*58 530	*46 160	*15.38	*49.60	—	.058	.107	.40	075	—		b	
7	7' 10"	0°	*58 990	*39 500	*18.29	*61.43	—	052	110	.49	085	—		A, b	
6.3			58 084	39 678	17.58	55.21		061	109	.44	.078				
6.3	1' 9"	− 80°	*59 220	*44 350	*19.00	*50.40	—	068	112	.37	07	—		b	
7	3' 0"	180°	*61 480	*41 780	*14.17	*50.20	—	.057	094	45	.085	—		b	
8	3' 0"	− 10°	*61 130	*43 420	*13.96	*54.90	—	.060	.111	.49	.075	—	Bessemer Steel	b	6" Converse Joint 8.26 Lbs.
5.5	2' 1"	+ 10°	*59 710	*40 910	*11.58	*51.13	—	.061	.107	.45	.08	—		b	
6.5	7' 0"	180°	*56 350	*39 250	*16.04	*55.00	—	.069	103	47	.08	—		b	
6.7			59 578	41 942	14.75	52.33		063	.105	.45	.078				
8.2	2' 4"	−150°	*58 690	*44 280	*15.75	*53.70	—	076	.105	40	08	—		b	
7	18' 0"	+135°	*62 580	*45 010	*13.54	*53.60	—	.079	.111	.49	08	—		b	
8.3	5' 0"	−160°	*61 000	*46 270	*18.79	*55.80	—	080	.112	.36	08	—	Bessemer Steel	b	5⅝" Casing 10.46 Lbs
8	2' 4"	−160°	*64 530	*47 630	*14.54	*48.20	—	.071	.111	.40	075	—		b	
10	2' 8"	− 10°	*59 160	*42 320	*15.25	*56.80	—	.077	.108	.33	.07	—		b	
8.3			61 192	45 102	15.57	53.62		077	.109	.396	077				
7	2' 1"	+ 45°	*62 610	*44 390	*16.04	*55.20	—	081	.119	38	.07	—		b	
8	2' 6"	− 40°	*58 660	*43 640	*13.92	*52.20	—	.072	101	38	07	—		b	
8	2' 6"	+ 15°	*61 390	*46 010	*15.38	*53.80	—	.078	.108	.39	.07	—	Bessemer Steel	b	5⅝" Casing 10.46 Lbs.
8	3' 6"	+ 10°	*62 860	*44 400	*15.92	*50.70	—	.084	.112	.36	08	—		b	
8	2' 0"	+ 10°	*64 050	*48 200	*15.46	*49.90	—	.071	.107	.36	.07	—		b	
7.8			61 914	45 328	15 34	52.36		077	.109	374	072				
7	2' 0"	−150°	*64 320	*47 390	*14.25	*54.20	—	.084	111	39	07	—		b	
8	2' 2"	0°	*57 670	*42 560	*12.25	*64.40	—	.077	107	40	.07	—	Bessemer Steel	b	5⅝" Casing 10.46 Lbs
10	4 3"	0°	*63 000	*46 660	*18.25	*52.50	—	.074	109	35	07	—		b	
8.3			61 663	45 537	14.92	57.03		078	109	38	.07				
6	6' 0"	0°	*59 990	*43 650	*16.50	*52.80	—	085	106	35	.085	—		a	
6	17' 8"	−150°	*57 970	*38 930	*19.71	*56.70	—	066	098	42	.08	—		a	
5	2' 2"	− 20°	*63 660	*49 650	*13.84	*53.40	—	075	.106	40	075	—	Bessemer Steel	a	5⅝" Casing 10.46 Lbs.
7	2' 1"	180°	*57 060	*42 230	*19.38	*58.20	—	.062	086	47	.08	—		a	
5	15' 3"	0°	*62 150	*45 270	*15.63	*50.60	—	085	.110	.39	075	—		a	
5.8			60 166	43 946	17.01	54.34		.075	.101	.41	.079				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

* These tests were made on spec
and are therefore too high for Yield Point and
The following are average values for 30 and
Tensile Strength, lbs.
Yield Point, lbs. per sq
Elongation, % in 8 inch
Reduction of Area, %

Test Number	Outside Diameter Inches				Thickness of Wall Inches				Length of Tube Feet			Weight of Tube Lbs. per Foot		Collapsing Pressure			Collapse	
	Nominal	Average	At Place of Collapse		Nominal	Average	At Place of Collapse		As Reported	Actual	Unsupported			Pounds per Sq. Inch	Gage Used	Rate of Increase Lbs. per Sec.	Length	
			Greatest	Least			Greatest	Least				Nominal	Actual				In Feet	In Dia's.
240		6.035	6.050	5.980		0.179	0.196	0.141		20' 0¼"	19' 10½"		11.19	1095	C	3	4' 0"	8
241		6.021	6.075	5.970		0.219	0.240	0.198		20' 0¼"	19' 10½"		13.57	1600	C	1	3' 6"	7
242	6.000	6.035	6.040	5.990	0.220	0.185	0.197	0.158	20' 0"	20' 0¼"	19' 10½"	14.20	11.57	1272	C	3	5' 0"	10
243		6.034	6.080	5.990		0.170	0.190	0.144		20' 0¼"	19' 10½"		10.62	875	B	2	3' 6"	7
244		5.988	6.020	5.940		0.193	0.253	0.193		20' 0¼"	19' 10½"		11.97	1750	C	2	5' 0"	10
Average		6.023	6.053	5.974		0.189	0.215	0.167		20' 0¼"	19' 10½"		11.78	1318		2.2	4' 2"	8.4
245		6.010	6.060	5.980		0.193	0.252	0.174		20' 0¼"	19' 7"		11.97	1375	C	5	4' 0"	8
246		6.001	6.010	5.930		0.230	0.252	0.189		20' 0¼"	19' 8½"		14.17	1900	C	5	3' 6"	7
247	6.000	6.055	6.070	5.970	0.220	0.179	0.200	0.153	20' 0"	20' 0¼"	19' 8½"	14.20	11.22	1010	B	5	3' 0"	6
248		6.004	6.060	5.890		0.232	0.243	0.210		20' 0"	19' 8½"		14.30	1800	C	5	4' 6"	9
249		6.036	6.050	5.980		0.228	0.238	0.192		20' 0"	19' 8½"		14.15	1200	C	5	4' 6"	9
Average		6.021	6.050	5.950		0.212	0.237	0.184		20' 0¼"	19' 8½"		13.16	1457		5	3' 11"	7.8
250		6.016	6.030	5.990		0.192	0.201	0.174		20' 0"	19' 5⅜"		11.76	1450	C	2	5' 0"	10
251		6.016	6.050	5.980		0.214	0.233	0.173		20' 0¼"	19' 5½"		13.25	1750	C	5	5' 0"	10
252	6.000	6.020	6.050	5.960	0.203	0.222	0.235	0.204	20' 0"	20' 0"	19' 5⅜"	12.04	13.73	2075	C	5	6' 0"	12
253		6.020	6.060	5.950		0.184	0.248	0.164		20' 0¼"	19' 5¾"		11.45	950	B	2	4' 0"	8
254		6.005	6.070	5.950		0.220	0.251	0.201		20' 0¼"	19' 5½"		13.59	1550	C	2	5' 0"	10
Average		6.015	6.052	5.966		0.206	0.234	0.187		20' 0¼"	17' 5¾"		12.80	1555		3.2	5' 0"	10
255		6.028	6.050	5.990		0.192	0.208	0.170		20' 0"	19' 8½"		11.78	790	B	2	3' 6"	7
256		6.021	6.050	5.990		0.185	0.195	0.149		20' 0¼"	19' 8½"		11.55	1450	C	5	4' 0"	8
257	6.000	6.025	6.060	5.960	0.203	0.189	0.210	0.165	20' 0"	20' 0¼"	19' 8½"	12.04	11.77	1250	C	3	3' 0"	6
258		6.013	6.020	5.950		0.182	0.207	0.150		20' 0"	19' 8½"		11.30	1350	C	5	4' 0"	8
259		6.024	6.060	5.960		0.182	0.219	0.160		20' 0"	19' 8½"		11.35	1100	C	2	3' 0"	6
Average		6.022	6.048	5.970		0.186	0.208	0.159		20' 0¼"	19' 8½"		11.59	1188		3.4	3' 6"	7
260		6.054	6.180	5.920		0.260	0.286	0.242		20' 0¼"	19' 4½"		16.06	1755		2	4' 6"	9
261		6.027	6.040	5.770		0.251	0.296	0.218		20' 0¼"	19' 5½"		15.49	1750		2	6' 3"	12.5
262	6.000	6.033	——	——	0.271	0.263	——	——	20' 0"	20' 0"	19' 5¼"	16.70	16.20	——	C	—	——	——
263		6.025	6.060	5.960		0.280	0.300	0.265		20' 0¼"	19' 5¼"		17.17	2600		5	5' 0"	10
264		6.022	6.080	6.000		0.260	0.272	0.244		20' 0¼"	19' 5½"		15.99	2450		5	6' 0"	12
Average		6.032	6.090	5.968		0.263	0.289	0.242		20' 0¼"	19' 5⅜"		16.18	2139		3.5	5' 5"	10.9
273		6.021	6.050	5.960		0.270	0.289	0.240		20' 0¼"	19' 8½"		16.59	2270		2	3' 8"	7.3
276		6.033	6.060	6.000		0.259	0.290	0.226		20' 0¼"	19' 8½"		15.94	2475		5	4' 0"	8
279	6.000	6.072	6.100	6.000	0.271	0.254	0.304	0.217	20' 0"	20' 0¼"	19' 8½"	16.70	15.79	2360	C	5	3' 9"	7.5
282		6.024	6.060	5.970		0.267	0.288	0.230		20' 0¼"	19' 8½"		16.44	2250		5	1' 10"	3.7
285		6.022	6.050	5.980		0.270	0.322	0.222		20' 0"	19' 8½"		16.55	2550		5	5' 0"	10
Average		6.034	6.064	5.982		0.264	0.299	0.227		20' 0¼"	19' 8¾"		16.26	2381		4.4	3' 8"	7.3
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 35.—TABULAR STATEMENT OF PR

e on specimens cut from distorted portions of the tube,
Point and too low for Elongation and Reduction of Area
or 30 undistorted specimens from groups 200 and 300.
gth, lbs. per sq in. — — 57,890.
s. per sq in. — — — — 36,860
in 8 inches — — — — — — — 21.18
Area, % — — — — — — — 57.34

R - See General Remark Sheet Nº 33.
a & b - See types of Test Heads on General Remark Sheet Nº 33.
bp - Test Head "b", with plugs in ends of tube.

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

ollapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported.
gth In Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
8	2' 1"	− 90°	*60 830	*42 590	*14.59	*53.70	——	.070	.105	.33	.075	——		R, b	
7	2' 6"	0°	*61 430	*47 420	*18.25	*55.00	——	.049	.070	.36	.075	——		b	
10	3' 0"	0°	*59 170	*46 200	*10.96	*57.90	——	.069	.107	.32	.065	——	Bessemer Steel	b	5⅝" Casing 14.20 Lbs
7	2' 0"	+ 90°	*61 240	*46 360	*16.46	*56.80	——	.076	.106	.38	.075	——		b	
10	17' 6"	0°	*61 990	*41 710	*14.92	*48.80	——	.079	.101	.39	.07	——		b	
8.4			60 932	44 856	15.04	54.44		.068	.098	.36	.072				
8	17' 0"	− 20°	*61 260	*40 890	*15.58	*55.90	——	.080	.103	.40	.075	——		a	
7	2' 1"	+ 20°	*65 590	*47 750	*16.48	*51.40	——	.081	.103	.40	.07	——		a	
6	18' 0"	0°	*62 190	*46 690	*13.34	*54.10	——	.082	.106	.35	.075	——	Bessemer Steel	a	5⅝" Casing 14.20 Lbs
9	2' 3"	− 20°	*67 660	*49 150	*13.13	*50.60	——	.086	.107	.40	.08	——		a	
9	17' 0"	+ 15°	*65 850	*48 140	*16.67	*51.80	——	.094	.112	.35	.075	——		R, a	
7.8			64 510	46 924	15.04	52.76		.085	.106	.38	.075				
10	3' 6"	− 5°	*62 390	*46 000	*16.34	*55.27	——	.096	.102	.41	.08	——		bp	
10	3' 0"	− 10°	*64 090	*50 000	*14.25	*52.20	——	.099	.104	.37	.076	——		bp	
12	3' 8"	0°	*65 330	*46 810	*13.58	*49.90	——	.092	.100	.25	.07	——	Bessemer Steel	R, bp	5⅝" Casing 12.04 Lbs
8	7' 9"	0°	*61 030	*47 240	*13.50	*56.00	——	.054	.085	.40	.08	——		bp	
10	15' 6"	0°	*61 890	*47 550	*14.29	*50.40	——	.083	.113	.40	.07	——		bp	
10			62 946	47 520	14.39	52.75		.083	.101	.37	.075				
7	2' 0"	− 45°	*61 410	*42 920	*14.92	*56.10	——	.075	.107	.38	.07	——		a	
8	2' 0"	− 20°	*65 530	*48 290	*15.83	*56.90	——	.086	.107	.45	.075	——		a	
6	15' 9"	180°	*63 940	*46 440	*14.21	*55.20	——	.051	.080	.36	.07	——	Bessemer Steel	a	5⅝" Casing 12.04 Lbs
8	16' 10"	+ 10°	*59 930	*44 090	*13.88	*50.40	——	.085	.111	.35	.075	——		a	
6	3' 0"	+ 30°	*65 560	*45 470	*15.59	*52.50	——	.072	.109	.40	.085	——		a	
7			63 274	45 442	14.89	54.22		.074	.103	.39	.075				
9	17' 6"	− 10°	*60 860	*41 650	*19.13	*55.70	——	.078	.112	.35	.075	——		R, bp	
12.5	10' 3"	− 10°	57 360	41 400	13.38	57.70	——	.088	.100	.33	.07	——		R, bp	
——	——	——	59 150	37 480	22.92	59.50	——	.091	.102	.36	.08	——	Bessemer Steel	R, bp	5⅝" Casing 16.70 Lbs
10	16' 9"	− 5°	58 830	37 140	19.05	56.70	——	.094	.103	.47	.075	——		R, bp	
12	17' 0"	0°	58 050	36 810	13.82	60.50	——	.085	.104	.39	.07	——		bp	
10.9								.087	.104	.38	.074				
7.3	18' 0"	0°	*63 810	*47 510	*18.33	*58.40	——	.080	.101	.38	.07	——		a	
8	2' 4"	0°	*65 830	*49 570	*14.29	*54.00	——	.099	.109	.42	.08	——		a	
7.5	18' 0"	0°	57 110	37 120	19.29	58.30	——	.092	.103	.37	.07	——	Bessemer Steel	R, a	5⅝" Casing 16.70 Lbs
3.7	0' 11"	0°	*57 080	*36 030	*22.17	*58.70	——	.092	.106	.52	.085	——		R, a	
10	3' 3"	+ 20°	*63 640	*42 150	*16.33	*54.80	——	.075	.120	.34	.075	——		a	
7.3								.088	.108	.41	.076				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

OF PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

A - See
a b c -
* - Not

Test Number	Outside Diameter Inches: Nominal	Outside Diameter: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Thickness of Wall Inches: Nominal	Thickness of Wall: Average	At Place of Collapse: Greatest	At Place of Collapse: Least	Length of Tube Feet: As Reported	Length of Tube: Actual	Length of Tube: Unsupported	Weight of Tube Lbs per Feet: Nominal	Weight of Tube: Actual	Collapsing Pressure: Pounds per Sq. Inch	Gage Used	Rate of Increase Lbs per Sec	Collapsed Length: In Feet	Collapsed Length: In Dia's.
300		6.657	6.675	6.505		0.157	0.185	0.152		20' 0¼"	19' 7⅞"		10.87	710		3	2' 0"	3.6
301		6.657	6.680	6.595		0.165	0.188	0.145		20' 0⅜"	19' 9¼"		11.47	720		3	2' 8"	4.8
302	6.625	6.653	6.705	6.570	0.172	0.168	0.189	0.153	20' 0"	20' 0"	19' 9⅝"	11.58	11.62	630	B	3	2' 0"	3.6
303		6.653	6.685	6.575		0.165	0.205	0.145		20' 0"	19' 8⅞"		11.35	730		5	3' 0"	5.4
304		6.652	6.685	6.600		0.164	0.214	0.123		20' 0¼"	19' 9⅜"		11.42	600		3	3' 0"	5.4
Average		6.654	6.686	6.569		0.164	0.196	0.144		20' 0⅛"	19' 8 13/16"		11.35	678		3.4	2' 6"	4.6
305		6.680	6.725	6.605		0.196	0.220	0.180		20' 0⅜"	19' 8⅞"		13.57	1100		3	2' 0"	3.6
306		6.693	6.700	6.590		0.200	0.207	0.194		20' 0"	19' 9⅛"		13.85	1205		4	3' 0"	5.4
307	6.625	6.682	6.705	6.565	0.203	0.202	0.227	0.186	20' 0"	20' 0⅜"	19' 8⅜"	13.32	13.99	1275	C	5	2' 6"	4.5
308		6.687	6.670	6.575		0.205	0.225	0.200		20' 0⅜"	19' [illegible]		14.17	1265		5	3' 0"	5.4
309		6.676	6.675	6.570		0.196	——	——		20' 0"	19' 7⅞"		13.57	1075		4	3' 0"	5.4
Average		6.684	6.695	6.581		0.200	0.220	0.190		20' 0¼"	19' 7 15/16"		13.83	1184		4.2	2' 8"	4.9
310		6.661	6.640	6.610		0.260	0.278	0.245		20' 0¼"	19' 8⅜"		17.74	2275		5	3' 0"	5.4
311		6.674	6.655	6.605		0.254	0.277	0.225		20' 0⅜"	19' 6⅜"		17.39	2160		5	4' 0"	7.3
312	6.625	6.669	6.645	6.530	0.238	0.252	0.264	0.224	20' 0"	20' 0"	19' 8 11/16"	17.02	17.25	1975	C	5	3' 0"	5.4
313		6.657	6.615	6.555		0.250	——	——		20' 0¼"	19' 8⅞"		17.09	1820		5	2' 6"	4.5
314		6.671	6.650	6.575		0.251	0.283	0.228		20' 0"	19' 10⅛"		17.20	2175		5	3' 0"	5.4
315		7.046	7.070	7.000		0.158	0.177	0.122		20' 0¼"	19' 9⅞"		11.61	570		4	2' 6"	4.3
316		7.040	7.075	6.975		0.158	0.177	0.136		20' 0⅜"	19' 9⅞"		11.58	550		4	2' 0"	3.4
317	7.000	7.043	7.090	6.980	0.180	0.154	0.170	0.128	20' 0"	20' 0¼"	19' 9⅝"	12.34	11.31	575	B	3	2' 0"	3.4
318		7.033	7.040	6.980		0.166	0.214	0.135		20' 0¼"	19' 9⅞"		12.13	580		3	2' 4"	4.0
319		7.059	7.075	6.995		0.163	0.217	0.130		20' 0¼"	19' 9⅞"		11.96	540		5	2' 0"	3.4
Average		7.044	7.070	6.990		0.160	0.191	0.130		20' 0¼"	19' 9¾"		11.72	563		3.8	2' 2"	3.7
320		7.050	7.085	6.990		0.245	0.257	0.225		20' 0¼"	19' 8⅜"		17.76	1775		4	2' 4"	4.0
321		7.050	7.070	6.980		0.241	0.273	0.220		20' 0"	19' 8⅝"		17.50	1575		4	2' 6"	4.3
322	7.000	7.057	7.080	6.980	0.248	0.233	0.243	0.208	20' 0"	20' 0"	19' 8⅛"	17.51	16.97	1525	C	5	2' 8"	4.6
323		7.045	7.090	6.980		0.245	0.270	0.220		20' 0¼"	19' 8⅞"		17.77	1675		5	3' 0"	5.1
324		7.047	7.080	6.990		0.248	0.261	0.206		20' 0¼"	19' 8⅞"		18.02	1850		5	2' 6"	4.3
Average		7.050	7.081	6.984		0.242	0.261	0.216		20' 0⅛"	19' 8⅛"		17.60	1680		4.6	2' 7"	4.5
400		6.663	6.750	6.560		0.154	0.159	0.110		20' 1⅛"	19' 9⅞"		10.68	*400		*10	*20' 0"	*36.2
401		6.658	6.730	6.580		0.157	0.190	0.130		20' 0⅜"	19' 8⅞"		10.87	585		3	4' 6"	8.2
402	6.625	6.665	6.730	6.580	0.150	0.152	0.162	0.125	20' 0"	20' 1½"	19' 10"	10.39	10.57	585	B	3	4' 9"	8.6
403		6.661	6.710	6.570		0.154	0.182	0.140		20' 0¾"	19' 8¾"		10.67	560		3	5' 6"	10.0
404		6.657	6.720	6.580		0.153	0.173	0.125		20' 0⅜"	19' 8⅞"		10.59	520		2	4' 6"	8.2
Average		6.661	6.728	6.574		0.154	0.173	0.126		20' 0¾"	19' 9"		10.68	563		2.8	4' 10"	8.8
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 36 —TABULAR STATEMENT OF PRINC

R - See remark on General Remark Sheet Nº 33
a b c - See types of Test Heads on General Remark Sheet Nº 33.
* - Not in average, pressure continued after collapse

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

…lapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported.
…h Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs. per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
3.6	2' 0"	+ 90°	55 540	36 310	21.67	57.20	—	.067	.107	.38	.07	—		a	
4.8	7' 6"	− 30°	57 730	36 530	21.63	56.20	—	.065	.104	.39	.075	—		a	
3.6	10' 6"	0°	58 340	37 890	21.88	60.20	—	.059	.110	.38	.08	—	Bessemer Steel	a	$6\frac{1}{4}$" Casing 11.58 Lbs.
5.4	2' 9"	−110°	57 170	34 330	23.04	59.90	—	.060	.090	.39	.07	—		a	
5.4	2' 6"	180°	58 310	37 520	22.59	56.60	—	.068	.108	.38	.075	—		a	
4.6			57 419	36 516	22.16	58.02		.064	.104	.38	.074				
3.6	2' 8"	0°	57 790	36 200	23.21	57.10	—	.068	.112	.37	.09	—		a	
5.4	2' 6"	0°	57 790	36 830	18.84	42.30	—	.051	.108	.39	.065	—		a	
4.5	2' 0"	+ 10°	57 070	33 720	22.42	60.50	—	.054	.103	.37	.075	—	Bessemer Steel	a	$6\frac{1}{4}$" Casing 13.32 Lbs.
5.4	17' 6"	0°	57 100	36 090	24.42	61.90	—	.064	.102	.39	.075	—		b	
5.4	2' 7"	0°	57 490	35 970	23.42	57.90	—	.061	.098	.39	.07	—		a	
4.9			57 848	35 602	22.46	55.94		.060	.105	.38	.075				
5.4	2' 6"	0°	57 590	35 930	24.13	59.90	—	.055	.100	.39	.07	—	Bess. Steel	R, a	
7.3	2' 0"	+ 10°	57 690	36 610	25.33	60.80	—	.067	.116	.36	.07	—	" "	R, a	
5.4	7' 6"	+ 15°	54 360	32 740	20.42	62.60	—	.071	.106	.37	.075	—	" "	a	$6\frac{1}{4}$" Casing 17.02 Lbs.
4.5	18' 0"	180°	47 130	33 220	13.09	23.20	—	.015	.186	Trace	Trace	—	Wro't Iron	R, a	
5.4	13' 6"	0°	57 110	36 480	21.79	60.40	—	.054	.100	.37	.07	—	Bess. Steel	a	
4.3	9' 4"	+ 15°	57 430	38 510	17.92	54.50	—	.053	.093	.34	.07	—		a	
3.4	8' 6"	0°	56 440	35 880	21.17	59.70	—	.057	.106	.35	.07	—		a	
3.4	15' 8"	0°	56 850	36 780	16.71	61.60	—	.056	.095	.40	.075	—	Bessemer Steel	a	$6\frac{5}{8}$" Casing 12.34 Lbs.
4.0	11' 0"	0°	59 520	41 140	22.42	52.90	—	.069	.104	.37	.07	—		a	
3.4	14' 6"	−120°	56 070	38 920	19.25	58.30	—	.059	.106	.36	.08	—		a	
3.7			57 262	38 246	19.49	57.40		.059	.101	.36	.073				
4.0	9' 9"	0°	59 860	37 280	21.96	51.30	—	.072	.110	.35	.075	—		R, a	
4.3	3' 0"	+ 15°	59 660	38 250	21.33	56.30	—	.065	.101	.38	.07	—		a	
4.6	5' 0"	0°	57 730	35 500	23.34	57.90	—	.061	.094	.38	.07	—	Bessemer Steel	a	$6\frac{5}{8}$" Casing 17.51 Lbs.
5.1	2' 4"	180°	60 210	33 910	23.50	56.10	—	.061	.102	.37	.075	—		a	
4.3	4' 4"	0°	57 720	35 730	24.71	57.30	—	.060	.100	.37	.07	—		R, a	
4.5			59 036	36 134	22.97	55.78		.064	.101	.37	.072				
36.2	15' 0"	[illegible]	58 260	46 835	13.21	51.10	—	.061	.089	.37	.07	—		R, c	
9.2	5' 3"	+150°	52 785	37 565	15.50	53.90	—	.058	.090	.33	.07	—		c	
8.6	7' 0"	+140°	57 415	37 025	21.00	57.10	—	.070	.105	.35	.07	—	Bessemer Steel	c	Special $6\frac{5}{8}$" O.D. 10.39 Lbs.
10.0	6' 0"	− 77°	58 385	38 035	21.72	60.10	—	.075	.115	.32	.07	—		R, c	
9.2	15' 0"	0°	56 445	35 695	20.46	54.20	—	.070	.107	.32	.07	—		c	
8.8			56 658	39 031	18.42	55.28		.067	.101	.34	.07				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form.

Test Number	Outside Diameter, Inches: Nominal	Outside Diameter: Average	Outside Diameter at Place of Collapse: Greatest	Outside Diameter at Place of Collapse: Least	Thickness of Wall, Inches: Nominal	Thickness of Wall: Average	Thickness at Place of Collapse: Greatest	Thickness at Place of Collapse: Least	Length of Tube, Feet: As Reported	Length of Tube: Actual	Length of Tube: Unsupported	Weight of Tube, Lbs per Foot: Nominal	Weight of Tube: Actual	Collapsing Pressure: Pounds per Sq. Inch	Collapsing Pressure: Gage Used	Collapsing Pressure: Rate of Increase Lbs per Sec.	Collapsed: Length In Feet	Collapsed: Length In Dia's.
405		6.650	6.700	6.570		0.262	0.291	0.215		20' 1 1/4"	19' 9 3/4"		17.85	2135		3	3' 6"	6.3
406		6.653	6.680	6.630		0.262	0.303	0.243		20' 1 1/4"	19' 9 3/4"		17.85	1975		4	4' 6"	8.1
407	6.625	6.656	6.700	6.610	0.280	0.278	0.290	0.249	20' 0"	20' 2"	19' 10 1/8"	18.76	18.94	2560	C	3	5' 6"	10.0
408		6.651	6.670	6.610		0.261	0.277	0.230		20' 1 1/2"	19' 9 3/8"		17.81	2060		3	4' 6"	8.1
409		6.664	6.690	6.600		0.284	0.325	0.272		20' 0 1/4"	19' 8 3/8"		19.35	2340		1	5' 0"	9.1
Average		6.655	6.688	6.608		0.269	0.297	0.242		20' 1 1/4"	19' 9 3/8"		18.36	2214		2.8	4' 7"	8.3
410		6.686	6.730	6.590		0.241	0.259	0.220		20' 1 1/4"	19' 9 3/8"		16.59	1680		5	4' 0"	7.3
411		6.684	6.700	6.630		0.252	0.297	0.217		20' 1 1/4"	19' 9 3/8"		17.24	1940		2	6' 0"	10.9
412	6.625	6.675	6.710	6.590	0.238	0.256	0.309	0.224	20' 0"	20' 1 1/4"	19' 9 3/8"	17.02	17.51	1780	C	3	5' 0"	9.1
413		6.679	6.710	6.610		0.248	0.283	0.223		20' 1 1/4"	19' 9 3/8"		16.99	1710		2	5' 6"	10.0
414		6.682	6.700	6.600		0.248	0.262	0.217		20' 1 1/4"	19' 9 3/8"		17.01	1615		4	6' 0"	10.9
Average		6.681	6.710	6.604		0.249	0.282	0.220		20' 1 1/4"	19' 9 3/8"		17.07	1745		3.2	5' 4"	9.6
415		6.072	6.100	6.030		0.250	0.279	0.203		20' 1 1/4"	19' 9 7/8"		15.53	2220		3	4' 6"	9.0
416		6.042	6.060	6.000		0.262	0.277	0.206		20' 1 3/8"	19' 9 5/8"		16.16	2565		2	4' 6"	9.0
417	6.000	6.042	6.080	6.000	0.271	0.271	0.307	0.222	20' 0"	20' 1 1/4"	19' 9 3/8"	16.70	16.71	2400	C	1	6' 0"	12.0
418		6.047	6.080	6.020		0.277	0.307	0.244		20' 1 3/8"	19' 11 3/8"		17.08	2870		5	6' 0"	12.0
419		6.044	6.060	6.010		0.271	0.313	0.193		20' 1 3/8"	19' 9 5/8"		16.73	2575		4	5' 0"	10.0
Average		6.049	6.076	6.012		0.266	0.297	0.214		20' 1 3/8"	19' 10 1/8"		16.44	2528		3	5' 2"	10.4
1		8.657	——	——		0.176	0.170	0.170		20.120	19.818		15.92	450		—	6' 0"	8.3
2		8.637	——	——		0.191	0.195	0.179		19.998	19.676		17.24	625		—	6' 3"	8.7
3	8.625	8.641	——	——	0.180	0.186	0.184	0.168	20' 0"	19.997	19.695	16.07	16.77	535	B	—	5' 6"	7.7
4		8.640	——	——		0.183	0.191	0.171		20.004	19.702		16.54	450		—	5' 6"	7.7
5		8.638	——	——		0.191	0.197	0.173		20.011	19.709		17.23	620		—	5' 0"	7.0
Average		8.643				0.185	0.191	0.172		20.026	19.724		16.74	536			5' 8"	7.9
26		8.604	8.610	8.580		0.219	0.230	0.210	19' 2"	19.168	18.866		19.57	870	B	—	5' 6"	7.7
27		8.628	8.640	8.610		0.233	0.239	0.186	13' 8"	13.675	13.373		20.95	1115	C	—	5' 6"	7.7
28	8.625	8.659	8.660	8.630	0.229	0.213	0.222	0.195	12' 11"	12.932	12.630	20.10	19.23	850	B	1.4	5' 6"	7.7
29		8.650	8.690	8.620		0.213	0.221	0.180	12' 8"	12.674	12.372		19.16	750	B	1.4	5' 6"	7.7
30		8.670	8.690	8.660		0.197	0.211	0.175	12' 3"	12.252	11.950		17.79	650	B	1.4	5' 6"	7.7
Average		8.642	8.658	8.620		0.215	0.225	0.189					19.34	847		1.4	5' 6"	7.7
50		8.660	8.685	8.585		0.271	0.287	0.265		20.000	19.646		24.28	1435		4.9	5' 6"	7.7
51		8.688	8.715	8.605		0.274	0.280	0.266		19.988	19.634		24.54	1430		6.2	6' 0"	8.4
52	8.625	8.664	8.695	8.635	0.271	0.258	0.261	0.248	20' 0"	20.003	19.649	24.38	23.13	1320	C	3.1	5' 9"	8.0
53		8.660	8.675	8.625		0.272	0.282	0.255		19.992	19.638		24.32	1520		4.1	5' 6"	7.7
54		8.660	8.665	8.635		0.262	0.280	0.255		19.993	19.639		23.52	1485		5.2	6' 0"	8.4
Average		8.666	8.687	8.617		0.267	0.278	0.258		19.995	19.641		23.76	1438		4.7	5' 9"	8.0
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 37.—TABULAR STATEMENT OF PRINC

E - See type of Test Head on General Remark Sheet No. 33.

PRINCIPAL RESULTS OF COLLAPSING TESTS ON
NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES
CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

...lapsed Portion			Physical Properties				Chemical Analysis %								
...h Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs. per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide	Material	Remarks	Commercial Designation Tube as Reported
6.3	2' 0"	180°	56 415	36 430	20.54	44.20	—	.058	.107	.31	.07	—	Bessemer Steel	E	6" Full Weight 18.76 Lbs.
8.1	15' 0"	0°	57 340	37 120	19.46	55.10	—	.052	.107	.36	.075	—		E	
0.0	2' 0"	−154°	57 925	34 455	25.21	60.50	—	.038	.085	.33	.07	—		E	
8.1	18' 0"	+ 39°	55 525	34 845	19.08	42.50	—	.040	.100	.32	.07	—		E	
9.1	17' 0"	+ 13°	58 665	36 195	25.33	58.30	—	.068	.113	.36	.075	—		E	
8.3			57 174	35 809	21.92	52.12		.051	.102	.34	.072				
7.3	10' 0"	+ 50°	55 980	38 790	21.25	53.10	—	.079	.105	.32	.07	—	Bessemer Steel	E	6¼" Casing 17.02 Lbs.
0.9	8' 0"	− 43°	57 380	35 665	21.84	58.80	—	.074	.102	.36	.075	—		E	
9.1	6' 0"	−160°	58 395	39 375	24.46	59.50	—	.077	.100	.32	.07	—		E	
0.0	7' 6"	+ 45°	56 715	36 830	20.38	57.30	—	.072	.097	.31	.07	—		E	
0.9	16' 6"	+ 70°	55 775	38 780	20.25	61.00	—	.080	.103	.32	.07	—		E	
9.6			56 849	37 928	21.64	57.94		.076	.101	.33	.071				
9.0	18' 0"	− 20°	55 805	36 865	19.13	60.90	—	.082	.099	.34	.075	—	Bessemer Steel	E	5⅝" Casing 16.70 Lbs.
9.0	13' 0"	+ 75°	57 235	38 255	21.50	61.70	—	.076	.108	.35	.07	—		E	
2.0	16' 6"	+ 7½°	58 245	38 165	22.83	59.80	—	.083	.104	.32	.07	—		E	
2.0	3' 0"	−170°	56 720	36 005	22.38	56.10	—	.085	.106	.35	.07	—		E	
0.0	7' 0"	− 85°	58 645	39 675	19.08	57.80	—	.090	.105	.29	.07	—		E	
0.4			57 330	37 793	20.98	59.26		.083	.104	.33	.071				
8.3	11' 0"	− 18°	58 410	35 670	26.13	57.80	.005	.069	.109	.35	.08	—	Bessemer Steel	These Tests copied from Series I.	8¼" Casing 16.07 Lbs.
8.7	15' 6"	− 30°	60 490	38 030	21.04	57.20	.006	.077	.118	.32	.075	—			
7.7	4' 3"	+120°	60 020	36 420	24.00	58.73	.006	.077	.115	.31	.075	—			
7.7	15' 3"	− 30°	58 640	35 510	21.92	57.43	.006	.079	.104	.32	.08	—			
7.0	4' 9"	+ 15°	59 160	36 350	23.00	57.50	.010	.067	.106	.31	.075	—			
7.7			59 344	36 396	23.22	58.53	.007	.074	.110	.32	.077				
7.7	14' 10"	− 58°	56 700	34 060	22.79	59.40	.006	.068	.105	.38	.07	—	Bessemer Steel	These Tests copied from Series I	8¼" Casing 20.10 Lbs.
7.7	10' 1"	− 9°	57 770	34 730	22.33	53.70	—	.070	.102	.32	.075	—			
7.7	8' 2"	− 67°	60 530	37 700	16.67	57.20	—	.083	.117	.35	.08	—			
7.7	9' 1"	−145°	61 170	37 300	23.67	59.70	—	.069	.111	.45	.08	—			
7.7	4' 2"	− 58°	60 850	37 560	20.71	55.70	.006	.068	.118	.39	.075	—			
7.7			59 404	36 270	21.23	57.14		.072	.109	.38	.076				
7.7	16' 10"	+ 22°	58 820	34 920	23.75	57.90	.006	.070	.112	.35	.08	—	Bessemer Steel	These Tests copied from Series I	8¼" Casing 24.38 Lbs.
1.4	8' 2"	−150°	58 890	37 300	20.92	56.70	—	.084	.105	.35	.075	—			
1.0	16' 6"	−120°	58 030	37 180	22.79	56.40	.005	.072	.111	.32	.07	—			
7.7	17' 3"	− 68°	58 650	34 300	24.13	57.80	—	.085	.107	.27	.08	—			
1.4	7' 5"	−150°	57 500	35 800	20.71	57.50	—	.073	.109	.38	.08	—			
1.0			58 378	35 900	22.46	57.66		.077	.109	.33	.077				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 — SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

R· S
c· S

Test Number	Outside Diameter, Inches: Nominal	Outside Diameter: Average	Outside Diameter: At Place of Collapse, Greatest	Outside Diameter: At Place of Collapse, Least	Thickness of Wall, Inches: Nominal	Thickness of Wall: Average	Thickness of Wall: At Place of Collapse, Greatest	Thickness of Wall: At Place of Collapse, Least	Length of Tube, Feet: As Reported	Length of Tube: Actual	Length of Tube: Unsupported	Weight of Tube, Lbs per Foot: Nominal	Weight of Tube: Actual	Collapsing Pressure: Pounds per Sq Inch	Collapsing Pressure: Gage Used	Collapsing Pressure: Rate of Increase Lbs per Sec	Collapse Length: In Feet	Collapse Length: In Dia's
75		8.640	8.675	8.405		0.274	0.282	0.260	20' 0"	20.003	19.649		24.44	1375		4.1	6' 3"	8.7
76		8.663	8.685	8.455		0.272	0.292	0.271	20' 0"	19.988	19.634		24.38	1410		3.4	—	—
77	8.625	8.660	8.665	8.605	0.281	0.274	0.311	0.262	19' 10"	19.855	19.501	25.00	24.47	1275	C	7.3	7' 9"	10.8
78		8.660	8.700	8.595		0.280	0.294	0.274	18' 9"	18.750	18.396		25.08	1250		9.5	7' 0"	9.7
79		8.664	8.670	8.650		0.266	0.280	0.246	18' 4"	18.340	17.986		23.84	1425		8.5	6' 6"	9.0
98		8.666	8.680	8.645		0.308	0.312	0.276		19.990	19.636		27.44	1735		5.6	6' 3"	8.7
99		8.663	8.700	8.620		0.264	0.287	0.258		20.000	19.646		23.66	1375		5.4	—	—
100	8.625	8.646	8.665	8.615	0.322	0.297	0.318	0.300	20' 0"	19.999	19.645	28.177	26.44	1710	C	6.3	7' 0"	9.7
101		8.647	8.705	8.565		0.294	0.297	0.252		19.995	19.641		26.27	1650		7.0	6' 0"	8.4
102		8.656	8.675	8.620		0.303	0.312	0.286		20.012	19.658		27.00	1960		5.4	5' 9"	8.0
420		8.658	8.700	8.610		0.310	0.350	0.269		20' 1⅛"	19' 10½"		27.64	1805		4	5' 0"	7.0
421		8.666	8.690	8.630		0.303	0.360	0.239		20' 0½"	19' 9¾"		27.09	1935		5	5' 6"	7.7
422	8.625	8.672	8.730	8.600	0.322	0.307	0.403	0.277	20' 0"	20' 0⅝"	19' 9¼"	28.18	27.39	1635	C	3	6' 0"	8.3
423		8.649	8.710	8.610		0.303	0.325	0.255		20' 0⅜"	19' 9½"		27.00	1830		4	6' 0"	8.3
424		8.671	8.730	8.570		0.302	0.328	0.259		20' 2"	19' 10⅞"		26.98	1575		3	6' 0"	8.3
Average		8.663	8.712	8.604		0.305	0.353	0.259		20' 1⅛"	19' 9¾"		27.22	1756		3.8	5' 8"	7.9
425		8.666	8.680	8.630		0.349	0.387	0.321		20' 0¾"	—		31.00	2180		2	5' 0"	7.0
426		8.672	8.710	8.600		0.364	0.405	0.340		20' 1⅛"	19' 10½"		32.30	2155		2	5' 0"	7.0
427	8.625	8.678	8.710	8.650	0.363	0.346	0.395	0.343	20' 0"	20' 0¼"	19' 8⅞"	32.00	30.82	1830	C	3	5' 6"	7.7
428		8.664	8.710	8.610		0.353	0.420	0.317		20' 1⅛"	19' 10¼"		31.36	2045		3	4' 0"	5.6
429		8.683	8.730	8.630		0.356	—	—		20' 0¼"	19' 8¾"		31.62	1930		4	5' 6"	7.7
Average		8.673	8.708	8.624		0.354	0.402	0.330		20' 1"	19' 9⅞"		31.42	2028		2.8	5' 0"	7.0
430		6.984	7.020	6.950		0.290	0.323	0.264		20' 2¼"	19' 10⅞"		20.74	2445		4	4' 6"	7.7
431		6.975	7.000	6.920		0.283	0.322	0.247		20' 2"	—		20.23	2300		1	5' 6"	9.4
432	7.000	7.018	7.060	6.980	0.280	0.269	0.346	0.225	20' 2"	20' 2¼"	19' 10⅞"	20.12	19.33	1835	C	3	4' 6"	7.7
433		6.974	7.000	6.920		0.287	0.325	0.255		20' 2⅜"	19' 10⅞"		20.47	2180		2	6' 0"	10.0
434		6.984	7.020	6.940		0.269	0.320	0.235		20' 2"	19' 10½"		19.34	1975		4	4' 0"	6.9
Average		6.987	7.020	6.942		0.279	0.327	0.245		20' 2¼"	19' 10⅝"		20.02	2147		2.8	4' 11"	8.3
435		7.008	7.050	6.910		0.161	0.175	0.148		20' 2"	19' 10¼"		11.75	665		4	4' 6"	7.7
436		7.015	7.060	6.970		0.168	0.262	0.143		20' 2"	19' 10¼"		12.25	645		5	5' 6"	9.4
437	7.000	7.009	7.060	6.970	0.150	0.155	0.182	0.136	20' 2"	20' 2"	19' 10¼"	10.65	11.31	605	B	2	4' 6"	7.7
438		7.015	7.070	6.970		0.162	0.185	0.135		20' 2"	19' 10¼"		11.85	675		5	5' 0"	8.6
439		7.009	7.100	6.930		0.153	0.173	0.136		20' 2"	19' 10¼"		11.21	515		5	4' 0"	6.9
Average		7.011	7.068	6.950		0.160	0.175	0.140		20' 2"	19' 10¼"		11.67	621		4.2	4' 8"	8.1
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 38.—TABULAR STATEMENT OF PRINC

REID T. STEWART.

R - See remark on General Remark Sheet Nº 33

C - See type of Test Head on General Remark Sheet Nº 33

PRINCIPAL RESULTS OF COLLAPSING TESTS ON
NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

Collapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported.
...gth In Dia's	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq In	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos	Mang	Carbon	Oxide			
8.7	14' 2"	+ 68°	55 680	32 880	24.54	61.00	.010	.061	109	35	08	—	Bess Steel		
—	14' 11"	+ 40°	59 290	35 530	18.63	58.00	—	.065	106	40	08	—	" "	These Tests	
10.8	9' 5"	-120°	43 740	25 240	15.21	28.30	—	.019	147	Trace	Trace	1.68	Wro't Iron	copied from	8" Line Pipe 25.00 Lbs.
9.7	9' 3"	+ 92°	45 510	29 520	13.21	25.30	.054	.038	110	11	.065	2.58	" "	Series 1	
9.0	15' 3"	-135°	46 430	26 710	15.92	26.80	.062	.040	109	11	.065	1.72	" "		
8.7	14' 10"	-110°	55 680	28 830	27.83	65.60	—	.020	.022	.40	14	—	O H Steel		
—	2' 7"	180°	58 010	33 630	24.46	60.20	—	.059	113	.35	.075	—	Bess "	These Tests	
9.7	9' 2"	+ 18°	56 350	34 510	23.75	57.60	—	.067	112	33	.08	—	" "	copied from	8" Line Pipe 28.177 Lbs
8.4	17' 4"	+112°	58 300	35 590	24.54	59.80	—	.074	105	.35	.07	—	" "	Series 2	
8.0	13' 2"	-160°	60 130	37 490	23.67	56.90	—	.065	.102	40	.08	—	" "		
7.0	18' 0"	+ 25°	56 625	33 760	21.88	61.50	—	.064	109	34	.07	—		C	
7.7	14' 6"	+ 20°	56 405	35 110	23.00	59.30	—	.062	107	32	.075	—		C	
8.3	11' 6"	+ 50°	58 115	35 725	21.75	58.00	—	.060	.104	.33	.07	—	Bessemer Steel	C	8" Full Weight 28.18 Lbs
8.3	11' 0"	- 25°	58 785	37 365	23.33	57.20	—	.070	.107	.34	.07	—		C	
8.3	11' 0"	- 25°	57 215	35 760	21.08	57.20	—	.061	.116	.34	.07	—		C	
7.9			57 469	35 784	22.21	56.64		.063	.109	33	.071				
7.0	2' 0"	- 45°	44 770	31 545	8.88	24.80	—	.020	.286	Trace	Trace	—		C	
7.8	18' 0"	- 85°	50 645	33 825	15.84	27.00	—	.022	.226	Trace	Trace	—		R, C	
7.7	17' 0"	-135°	45 365	29 200	18.75	25.70	—	.018	109	Trace	Trace	—	Wrought Iron	C	8" Oil Well Tubing 32.00 Lbs.
5.6	18' 4"	-105°	39 660	29 705	7.25	22.80	—	.020	.256	Trace	Trace	—		C	
7.7	15' 6"	+ 85°	44 730	28 770	19.92	33.60	—	.017	.163	Trace	Trace	—		C	
7.0			45 034	30 609	14.13	26.78		.019	.208	Trace	Trace				
7.7	10' 6"	0°	63 915	41 585	24.83	54.00	—	.052	124	.30	.07	—		C	
9.4	17' 0"	0°	57 965	34 845	25.59	57.60	—	.057	107	.30	.07	—		C	
7.7	17' 0"	+110°	57 540	35 575	22.08	59.40	—	.063	102	.32	.07	—	Bessemer Steel	C	Special 7" O.D. 20.12 Lbs
10.0	9' 0"	- 10°	59 955	35 600	24.83	58.30	—	.066	116	32	.07	—		C	
6.9	18' 0"	+170°	60 740	38 480	23.96	58.50	—	.054	.136	27	.075	—		C	
8.3			60 023	37 217	24.26	57.56		.058	.117	31	.071				
7.7	14' 4"	+130°	60 445	39 580	20.83	52.30	—	.077	.126	.31	.07	—		C	
9.4	4' 0"	- 25°	58 425	37 805	21.13	57.00	—	.080	.111	.33	.07	—		C	
7.7	3' 0"	+120°	59 995	38 420	20.08	51.90	—	.076	.122	.34	.075	—	Bessemer Steel	C	7" Converse Joint 10.65 Lbs.
8.6	17' 6"	- 70°	58 715	36 900	23.00	53.30	—	.060	.104	.32	.07	—		C	
6.9	16' 3"	- 60°	69 200	39 675	20.46	56.00	—	.063	.104	.33	.075	—		C	
8.1			59 356	38 480	21.10	54.10		.071	.113	.33	.072				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

R- See rem…
Ckd- See …
* Laminate…

Test Number	Outside Diameter, Inches: Nominal	Average	At Place of Collapse: Greatest	Least	Thickness of Wall, Inches: Nominal	Average	At Place of Collapse: Greatest	Least
440	6.000	5.989	6.040	5.960	0.280	0.273	0.296	0.244
441		5.993	6.060	5.930		0.269	0.274	0.249
442		5.990	6.020	5.960		0.269	0.281	0.252
443		5.998	6.060	5.950		0.271	0.285	0.250
444		5.993	6.030	5.950		0.274	0.291	0.253
Average		5.993	6.042	5.950		0.271	0.285	0.250
445	10.000 *	10.037	10.150	9.910	0.156	0.167	0.250	0.147
446		10.031	10.120	9.900		0.167	0.204	0.153
447		10.045	10.140	9.940		0.166	0.194	0.143
448		10.035	10.150	9.900		0.170	0.194	0.150
449		10.055	10.180	9.950		0.157	0.182	0.150
Average		10.041	10.148	9.920		0.165	0.205	0.149
450	10.000	10.027	10.140	9.850	0.203	0.206	0.232	0.187
451		10.029	10.120	9.920		0.194	0.210	0.134
452		10.005	10.120	9.870		0.185	0.282	0.143
453		10.033	10.160	9.920		0.190	0.221	0.184
454		10.037	10.100	9.910		0.175	0.272	0.152
Average		10.026	10.128	9.894		0.194	0.243	0.160
455	10.000	10.000	10.100	9.900	0.300	0.319	0.356	0.301
456		9.990	10.080	9.900		0.312	0.350	0.303
457		9.989	10.130	9.850		0.317	0.363	0.304
458		10.003	10.100	9.920		0.317	0.356	0.310
459		10.023	10.090	9.890		0.314	0.381	0.297
Average		10.001	10.100	9.892		0.316	0.361	0.303
460	4.000	3.990	4.020	3.970	0.120	0.119	0.132	0.098
461		3.990	4.030	3.950		0.122	0.134	0.096
462		3.990	4.010	3.970		0.122	0.147	0.100
463		4.001	4.010	3.980		0.120	0.140	0.085
464		3.992	4.040	3.950		0.114	0.124	0.098
Average		3.993	4.022	3.964		0.119	0.135	0.095
465	4.000	4.010	4.020	3.980	0.180	0.173	0.203	0.140
466		4.014	4.050	3.990		0.178	0.277	0.158
467		4.012	4.050	3.960		0.173	0.200	0.170
468		4.018	4.050	4.010		0.184	0.192	0.165
469		4.017	4.040	4.000		0.169	——	——
Average		4.014	4.042	3.988		0.175	0.218	0.158
1	2	3	4	5	6	7	8	9

Test Number	Length of Tube, Feet: As Reported	Actual	Unsupported	Weight of Tube, Lbs per Foot: Nominal	Actual	Collapsing Pressure: Pounds per Sq Inch	Gage Used	Rate of Increase lbs per Sec	Collapsed …: Length In Feet	In Dia's
440	20' 0" to 20' 2"	20' 2[illegible]"	19' 10[illegible]"	17.12	16.68	2780	C	3	5' 6"	11
441		20' 2[illegible]"	19' 10[illegible]"		16.45	2150		4	5' 6"	11
442		20' 2[illegible]"	19' 10[illegible]"		16.45	2590		2	5' 0"	10
443		20' 2[illegible]"	19' 10[illegible]"		16.55	2460		4	6' 0"	12
444		20' 2[illegible]"	19' 10[illegible]"		16.73	2455		3	4' 0"	8
Average		20' 2[illegible]"	19' 10[illegible]"		16.57	2487		3.2	5' 2"	10.4
445	20' 0"	20' 2[illegible]"	19' 10[illegible]"	16.18	17.58	210	B	5	8' 6"	10.2
446		20' 2[illegible]"	19' 10[illegible]"		17.59	225		3	11' 0"	13.2
447		20' 2[illegible]"	19' 10[illegible]"		17.50	240		3	6' 0"	7.2
448		20' 2[illegible]"	19' 10[illegible]"		17.94	240		5	7' 0"	8.4
449		20' 2[illegible]"	19' 10[illegible]"		16.55	210		5	9' 0"	10.8
Average		20' 2[illegible]"	19' 10[illegible]"		17.43	225		4.2	8' 4"	10.0
450	20' 0"	20' 1[illegible]"	19' 10[illegible]"	21.00	21.57	425	B	5	7' 0"	8.4
451		20' 1[illegible]"	19' 9[illegible]"		20.35	390		5	8' 0"	9.6
452		20' 0[illegible]"	19' 9[illegible]"		19.43	305		4	11' 0"	13.2
453		20' 1[illegible]"	19' 9[illegible]"		19.94	395		5	6' 6"	7.8
454		20' 1[illegible]"	19' 9[illegible]"		20.54	400		5	8' 0"	9.6
Average		20' 1[illegible]"	19' 9[illegible]"		20.37	383		4.8	8' 1"	9.7
455	20' 0"	20' 2[illegible]"	19' 10[illegible]"	31.07	33.01	1280	C	4	6' 6"	7.8
456		20' 2[illegible]"	19' 10[illegible]"		32.27	1350		3	8' 0"	9.6
457		20' 2[illegible]"	19' 10[illegible]"		32.71	1275		4	7' 6"	9.0
458		20' 2[illegible]"	19' 10[illegible]"		32.81	1305		4	7' 0"	8.4
459		20' 2[illegible]"	19' 10[illegible]"		32.61	1385		4	8' 0"	9.6
Average		20' 2[illegible]"	19' 10[illegible]"		32.68	1319		3.8	7' 5"	8.9
460	20' 0"	20' 2"	19' 3[illegible]"	4.89	4.91	925	B	15	3' 0"	9.0
461		20' 2"	19' 4"		5.06	975	B	20	3' 0"	9.0
462		20' 2[illegible]"	19' 4"		5.03	1030	C	5	4' 3"	12.7
463		20' 2"	19' 4"		4.98	1030	C	5	3' 3"	9.8
464		20' 2[illegible]"	19' 3[illegible]"		4.71	860	B	3	4' 0"	12.0
Average		20' 2[illegible]"	19' 3[illegible]"		4.94	964		9.6	3' 6"	10.5
465	20' 0"	20' 2[illegible]"	19' 4"	7.35	7.08	2050	C	5	2' 8"	8.0
466		20' 2[illegible]"	19' 4"		7.28	2225		3	5' 8"	17.0
467		20' 2[illegible]"	19' 4"		7.11	2425		3	5' 0"	15.0
468		20' 2[illegible]"	19' 3[illegible]"		7.53	2540		3	6' 0"	18.0
469		20' 2[illegible]"	19' 3[illegible]"		6.94	2160		3	5' 9"	17.2
Average		20' 2[illegible]"	19' 3[illegible]"		7.19	2280		3.4	5' 0"	15.0
1	10	11	12	13	14	15	16	17	18	19

FIG. 39.—TABULAR STATEMENT OF PRIN…

See remark on General Remark Sheet No 33.
d- See types of Test Heads on General Remark Sheet No 33
ominated, not in average

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4   F.P.K., 1905.

psed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported
Dia's	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq.In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
1	17' 3"	− 25°	57 007	35 360	22.75	59.00	——	.059	.097	.35	.075	——		R, c	
1	17' 0"	−145°	54 580	33 945	24.96	59.80	——	.061	.097	.35	.075	——		c	
0	18' 0"	−145°	55 695	35 945	21.42	63.50	——	.053	.097	.35	.075	——	Bessemer Steel	c	Special 6" O.D. 17.12 Lbs
2	17' 0"	0°	58 420	36 525	24.04	59.40	——	.060	.103	.33	.07	——		c	
8	18' 0"	− 20°	57 105	33 570	25.96	59.60	——	.055	.098	.33	.07	——		R, c	
0.4			56 561	35 069	23.83	60.26		.058	.098	.34	.073				
0 2	4' 0"	− 60°	57 425	35 090	18.46	52.20	——	.059	.104	.42	.07	——		R, c	
3.2	14' 0"	+ 22½°	55 895	38 130	20.72	59.60	——	.057	.110	.33	.075	——		R, c	
7.2	7' 0"	−110°	55 405	37 975	20.58	62.60	——	.053	.097	.32	.07	——	Bessemer Steel	R, c	10" Converse Joint 16.18 Lbs
8.4	15' 6"	0°	56 245	37 895	19.71	56.10	——	.046	.090	.33	.07	——		R, c	
0 8	5' 0"	+ 35°	57 415	36 975	19.54	59.10	——	.063	.110	.34	.07	——		R, c	
0.0			56 477	37 213	19.84	57.92		.056	.102	.35	.071				
8 4	3' 0"	+ 45°	56 695	39 105	21.96	57.70	——	.052	.103	.30	.07	——		R, c	
9.6	4' 0"	− 45°	56 635	38 740	19.75	56.80	——	.081	.113	.33	.07	——		R, c	
3.2	12' 0"	−165°	58 880	40 985	18.38	53.10	——	.080	.118	.29	.075	——	Bessemer Steel	c	10" Boiler Tubing 21.00 Lbs
7.8	5' 0"	− 55°	53 675	36 455	15.58	58.50	——	.087	.104	.31	.075	——		R, c	
9.6	17' 0"	− 60°	53 865	38 300	22.21	55.40	——	.091	.100	.28	.07	——		R, c	
9.7			55 950	38 717	19.58	56.30		.078	.108	.30	.072				
7 8	17' 0"	+ 60°	59 145	35 505	22.38	54.80	——	.054	.097	.33	.07	——		c	
9 6	16' 6"	− 20°	57 065	36 135	23.75	55.90	——	.063	.102	.38	.07	——		c	
9 0	15' 6"	+ 25°	60 360	38 575	22.42	56.70	——	.057	.105	.35	.07	——	Bessemer Steel	c	Special 10" O.D. 31.07 Lbs
8.4	2' 6"	− 25°	57 440	33 455	23.50	58.80	——	.065	.108	.32	.07	——		c	
9.6	5' 0"	−120°	58 735	35 985	21.67	58.50	——	.053	.109	.34	.075	——		c	
8 9			58 549	35 935	22.74	56.94		.058	.104	.34	.071				
9 0	5' 0"	Weld not	57 445	39 290	18.08	55.10	——	.041	.103	.30	.07	——		R, d	
9.0	4' 6"	found	61 005	44 770	17.79	50.10	——	.049	.102	.33	.07	——		R, d	
2 7	18' 0"	−145°	58 380	39 150	23.46	57.60	——	.043	.078	.38	.075	——	Bessemer Steel	c	4" Converse Joint 4.89 Lbs
9 8	16' 10"	Weld not	54 240	37 245	18.13	56.00	——	.062	.100	.32	.07	——		d	
12.0	2' 0"	found	61 480	45 405	17.75	53.70	——	.063	.111	.35	.07	——		c	
10.5			58 510	41 172	19.04	54.50		.052	.099	.34	.071				
8 0	18' 7"	+ 25°	*43 785	*34 105	*10.25	*37.30	——	.092	.144	.40	.07	——		c	
17 0	17' 0"	+ 50°	58 675	38 075	23.83	57.00	——	.064	.100	.32	.07	——		c	
15.0	3' 0"	180°	56 005	40 100	19.00	49.10	——	.098	.103	.32	.07	——	Bessemer Steel	c	3½" English 7.35 Lbs
18.0	16' 0"	− 60°	61 965	39 965	21.21	53.30	——	.086	.172	.29	.075	——		c	
17.2	17' 0"	−125°	——	——	——	——	——	——	——	——	——	——		c	
15.0			58 882	39 380	21.35	53.13		.085	.130	.33	.071				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

R- See remarks
d- See type of
* Not in averag

Test Number	Outside Diameter Inches Nominal	Outside Diameter Average	At Place of Collapse Greatest	At Place of Collapse Least	Thickness of Wall Inches Nominal	Thickness of Wall Average	At Place of Collapse Greatest	At Place of Collapse Least	Length of Tube Feet As Reported	Actual	Unsupported	Weight of Tube Lbs per Foot Nominal	Weight of Tube Actual	Collapsing Pressure Pounds per Sq Inch	Gage Used	Rate of Increase Lbs per Sec	Collapsed Length In Feet	Length In Dia's.
470		4.029	4.080	3.970		0.215	0.227	0.189		20' 2¼"	19' 3¾"		8.77	3125		3	3' 0"	9
471		4.024	4.030	4.010		0.213	0.245	0.191		20' 2¼"	19' 3¾"		8.67	3125		2	2' 9"	8.3
472	4.000	4.029	4.050	3.970	0.226	0.210	0.243	0.200	20' 0"	20' 2¼"	19' 3¾"	9.00	8.57	3150	D	5	3' 0"	9
473		4.028	4.040	4.020		0.217	0.252	0.214		20' 2¼"	19' 3¾"		8.82	3375		2	2' 9"	8.3
474		4.019	4.040	3.990		0.205	0.224	0.175		20' 2¼"	19' 4"		8.32	3075		3	3' 0"	9
Average		4.026	4.048	4.000		0.212	0.238	0.194		20' 2¼"	19' 3 13/16"		8.63	3170		3	2' 11"	8.7
475		4.020	4.030	4.010		0.324	0.380	0.296		20' 2"	19' 4"		12.79	5525		10	2' 6"	7.5
476		4.012	4.040	3.980		0.332	0.393	0.306		20' 0½"	19' 2"		13.05	5625		10	2' 6"	7.5
477	4.000	4.016	4.030	3.970	0.321	0.326	———	———	20' 0"	20' 0"	19' 1⅛"	12.47	12.85	5625	D	10	2' 6"	7.5
478		4.011	4.020	3.970		0.326	0.369	0.300		20' 1⅛"	19' 3⅛"		12.85	5600		10	2' 3"	6.8
479		4.010	4.020	3.970		0.328	0.370	0.300		20' 2"	19' 3⅛"		12.89	5425		10	2' 3"	6.8
Average		4.014	4.028	3.972		0.327	0.378	0.301		20' 1¼"	19' 2⅝"		12.89	5560		10	2' 5"	7.2
480		2.999	3.020	2.980		0.110	0.110	0.090		20' 0"	19' 1⅛"		3.40	1550		5	3' 0"	12
481		3.001	3.010	2.990		0.103	0.130	0.088		20' 0"	19' 1⅛"		3.18	* 465		5	3' 0"	12
482	3.000	3.006	3.030	2.940	0.109	0.110	0.118	0.096	20' 0"	19' 11⅞"	19' 1⅛"	3.33	3.40	1630	C	5	2' 3"	9
483		3.001	3.010	2.960		0.110	0.119	0.099		20' 0"	19' 1⅛"		3.40	1725		5	1' 6"	6
484		2.993	3.000	2.960		0.111	0.118	0.098		20' 0"	19' 1⅛"		3.425	2025		5	3' 9"	15
Average		3.000	3.014	2.966		0.109	0.119	0.094		20' 0"	19' 1⅛"		3.36	1733		5	2' 8"	10.8
485		2.986	3.010	2.970		0.112	0.124	0.098		20' 0"	19' 1⅛"		3.45	1800		5	4' 0"	16
486		2.997	3.020	2.970		0.112	0.120	0.092		19' 11⅞"	19' 1½"		3.45	1850		5	3' 0"	12
487	3.000	2.998	3.020	2.980	0.120	0.112	0.130	0.091	20' 0"	19' 11⅞"	19' 1⅛"	3.63	3.45	1960	C	5	3' 3"	13
488		2.998	3.020	2.970		0.114	0.115	0.100		19' 11⅞"	19' 1⅛"		3.53	2025		5	3' 6"	14
489		2.992	3.000	2.960		0.113	0.123	0.104		20' 0"	19' 1⅛"		3.48	2175		10	3' 9"	15
Average		2.994	3.014	2.970		0.113	0.122	0.097		19' 11⅞"	19' 1⅛"		3.47	1962		6	3' 6"	14
490	3.000	2.997	3.010	2.980	0.150	0.147	0.151	0.138	20' 0"	20' 1⅛"	19' 2⅝"	4.57	4.48	3350	D	3	4' 9"	19
491	3.000	2.987	3.010	2.970	0.150	0.139	0.139	0.125	20' 0"	20' 0"	19' 1⅛"	4.57	4.23	2575	-	20	2' 6"	10
Average		2.992	3.010	2.975		0.143	0.145	0.132		20' 0¾"	19' 2⅜"		4.36	2963			3' 8"	14.5
495		2.990	3.010	2.970		0.190	0.218	0.166		20' 0⅞"	19' 2½"		5.68	4200		3	2' 10½"	11.5
496		2.996	3.020	2.980		0.191	0.216	0.176		20' 1"	19' 2¾"		5.73	4200		8	3' 6"	14
497	3.000	2.997	3.020	2.960	0.180	0.190	0.215	0.161	20' 0"	20' 1"	19' 2¾"	5.42	5.68	4175	D	3	3' 3"	13
498		3.000	3.020	2.960		0.182	0.192	0.165		20' 1"	19' 2¾"		5.48	3700		5	3' 3"	13
499		2.994	3.000	2.980		0.189	0.227	0.170		20' 1¼"	19' 3"		5.65	4200		3	3' 6"	14
Average		2.995	3.014	2.970		0.188	0.214	0.168		20' 1"	19' 2¾"		5.64	4095		3.4	3' 3"	13.1
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19

FIG. 40.—TABULAR STATEMENT OF PRI

remarks on General Remark Sheet Nº 33.
type of Test Head on General Remark Sheet Nº 33
n average, valve between tank and gage almost closed

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF R.T STEWART, 1902-4 F.P.K., 1905

llapsed Portion			Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported
th n Dia's.	Distance from End	Angular Distance from Weld	Tensile Strength Lbs per Sq In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos	Mang.	Carbon	Oxide			
9	15' 3"	+140°	58 465	35 710	23.25	56.90	—	052	108	38	07	—		d	
8.3	18' 5"	−160°	59 145	36 835	24.54	59.80	—	050	103	33	09	—		d	
9	11' 0"	+110°	58 265	34 565	23.75	62.90	—	052	.096	.43	09	—	Bessemer Steel	d	3½" Full weight 9.00 Lbs
8.3	7' 4"	−130°	55 885	35 675	24.13	59.60	—	.044	099	.31	.075	—		d	
9	4' 8"	− 35°	60 280	36 125	22.88	56.70	—	.065	103	.41	085	—		d	
8.7			58 392	35 782	23.71	59.18		053	102	.37	.082				
7.5	1' 6"	−120°	59 950	38 640	22.17	59.30	—	059	.119	.30	07	—		d	
7.5	18' 6"	+ 25°	57 915	34 875	23.92	60.80	—	.052	.094	28	.085	—		d	
7.5	1' 6"	0°	60 175	40 875	21.13	56.20	—	.063	.090	.28	.08	—	Bessemer Steel	d	3½" Extra Strong 12.47 Lbs
6.8	1' 5"	+175°	58 740	37 860	22.88	58.20	—	.046	086	.31	.07	—		d	
6.8	1' 7"	−120°	56 455	35 255	24.83	62.50	—	.053	104	.33	.08	—		d	
7.2			58 647	37 601	22.99	59.40		055	.099	30	.077				
12	18' 0"	Weld not found	64 600	41 670	19.63	51.50	—	054	112	36	075	—		d	
12	18' 0"	+150°	—	—	—	—	—	—	—	—	—	—		R, d	
9	18' 5"	+170°	57 560	41 935	18.00	56.40	—	050	092	34	07	—	Bessemer Steel	d	3" Standard Boiler Tubing 3.33 Lbs
6	1' 6"	0°	61 160	39 595	20.79	56.60	—	046	.102	.33	.07	—		d	
15	1' 11"	+ 35°	60 305	42 955	20.50	52.10	—	063	.111	36	075	—		d	
10.8			60 906	41 539	19.73	54.15		.053	104	.35	073				
16	11' 0"	+ 30°	58 285	40 645	21.13	52.30	—	.046	088	.29	07	—		d	
12	1' 9"	Weld not found.	59 095	41 900	19.13	52.30	—	045	103	.30	.08	—		d	
13	2' 0"	+ 45°	64 065	45 170	18.92	51.70	—	.040	.095	31	075	—	Bessemer Steel	d	3" Locomotive Boiler Tubing 3.43 Lbs
14	2' 0"	−170°	62 455	43 355	19.92	50.60	—	053	112	29	08	—		d	
15	17' 10"	Weld not found.	59 195	42 015	18.54	57.20	—	065	.104	33	075	—		d	
14			60 619	42 617	19.53	52.82		.050	100	30	.076				
19	15' 0"	0°	59 705	40 785	20.42	50.00	—	076	110	29	075	—	Bess Steel	d	Special 3" O.D. 4.57 Lbs
10	13' 0"	Weld not found	45 705	31 025	23.08	56.90	—	025	041	Trace	07	—	" "	R, d	" " " " " "
14.5			52 705	35 905	21.75	53.45									
11.5	12' 3"	Weld not found.	70 315	45 350	21.25	51.60	—	071	099	.32	08	—		d	
14	10' 3"	+ 35°	57 455	37 755	20.42	57.50	—	.060	103	31	.08	—		d	
13	13' 0"	Weld not found.	59 215	37 330	22.25	56.80	—	060	100	.30	.075	—	Bessemer Steel	d	Special 3" O.D. 5.42 Lbs
13	16' 8"	+ 15°	58 375	36 255	19.42	57.90	—	.066	.102	.30	07	—		d	
14	10' 3"	Weld not found.	62 800	38 415	22.46	57.60	—	.065	.095	.29	07	—		d	
13.1			61 432	39 021	21.16	56.28		.064	..100	.30	.075				
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

F Principal Results of Tests, Series 2.

SERIES 2 — SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, *for lengths of 20 feet, between end connections tending to hold the tube to a circular form.*

Test Number	Outside Diameter, Inches: Nominal	Outside Diameter: Average	Outside Diameter at Place of Collapse: Greatest	Outside Diameter at Place of Collapse: Least	Thickness of Wall, Inches: Nominal	Thickness of Wall: Average	Thickness at Place of Collapse: Greatest	Thickness at Place of Collapse: Least	Length of Tube, Feet: As Reported	Length: Actual	Length: Unsupported	Weight of Tube, Lbs per Foot: Nominal	Weight: Actual	Collapsing Pressure: Pounds per Sq. Inch	Gage Used	Rate of Increase Lbs per Sec.	Coll… Length In Feet
500	10.750	10.776	10.830	10.740	0.500	0.508	0.570	0.479	15' 0" to 20' 0"	19' 11½"	19' 8"	54.25	55.666	2450	C	5	8' 6"
501		10.778	10.860	10.660		0.511	0.563	0.480		18' 5¼"	18' 2¼"		56.068	2575		4	8' 0"
502		10.767	——	——		0.508	0.571	0.490		19' 11¼"	19' 7⅞"		55.624	2620		3	——
503		10.788	10.850	10.710		0.506	0.524	0.486		19' 11⅞"	19' 7⅞"		55.595	2490		5	7' 3"
504		10.787	10.810	10.740		0.528	0.596	0.511		19' 5¼"	19' 1⅞"		57.866	2790		4	8' 0"
Average		10.779	10.838	10.713		0.512	0.565	0.489		19' 5⅞"	19' 3⅛"		56.164	2585		4.2	7' 11"
505	12.750	12.792	12.830	12.730	0.500	0.505	0.590	0.460	15' 0" to 20' 0"	19' 11⅞"	19' 8¼"	65.00	66.269	2385	C	5	10' 0"
506		12.772	12.920	12.710		0.516	0.599	0.489		19' 11⅞"	19' 8⅞"		67.535	2285		5	11' 0"
507		12.810	12.860	12.730		0.507	0.526	0.488		19' 11⅞"	19' 7⅞"		66.548	2090		3	10' 6"
508		12.794	12.850	12.750		0.515	0.552	0.502		19' 11⅞"	19' 7⅞"		67.526	2000		5	9' 0"
509		12.782	12.830	12.730		0.513	0.573	0.501		19' 11½"	19' 7⅞"		67.159	2220		5	9' 6"
Average		12.790	12.838	12.730		0.511	0.568	0.488		19' 11⅝"	19' 8"		67.007	2196		4.6	10' 0"
510	13.000	13.042	13.070	12.990	3½ to 4 B.W.G.	0.243	——	——	18' 0" to 20' 0"	20' 0"	19' 8½"	——	33.300	440	B	5	13' 0"
511		13.024	13.070	12.960		0.244	——	——		20' 0"	19' 8½"		33.250	430		4	11' 0"
512		13.038	13.050	12.980		0.244	——	——		20' 0"	19' 8½"		33.350	515		5	12' 0"
513		13.038	13.060	12.970		0.246	——	——		20' 0"	19' 8½"		33.600	480		5	11' 0"
514		13.038	13.080	13.020		0.245	——	——		20' 0"	19' 8½"		33.500	450		4	13' 0"
Average		13.036	13.066	12.784		0.244				20' 0"	19' 8½"		33.400	463		4.6	12' 0"
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18

FIG. 41.—TABULAR STATEMENT OF

R- See remark on General Remark Sheet Nº 33.
C- See type of Test Head on General Remark Sheet Nº 33.

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K., 1905.

Collapsed Portion				Physical Properties				Chemical Analysis %						Material	Remarks	Commercial Designation of Tube as Reported
Length		Distance from End	Angular Distance from Weld	Tensile Strength Lbs. per Sq. In.	Yield Point Pounds per Square Inch	Elongation % in 8 Inches	Reduction of Area %	Silicon	Sulphur	Phos.	Mang.	Carbon	Oxide			
...eet	In Dia's.															
6"	9.5	15' 7"	+165°	54 405	30 805	27.83	59.70	—	.050	088	.37	.07	—		C	
0"	8.9	14' 6"	+140°	56 260	37 210	18.88	60.40	—	.054	100	.36	.07	—		C	
—	—	—	—	56 060	31 245	25.42	59.40	—	.059	.087	.34	.07	—	Bessemer Steel	C	10" Extra Strong 54.25 Lbs.
3"	8.1	3' 8"	0°	57 595	33 780	22.92	42.10	—	.051	.105	.32	.075	—		C	
0"	8.9	15' 0"	0°	55 200	31 495	23.42	52.50	—	.058	.091	.34	.07	—		C	
11"	8.9			55 902	32 907	23.69	54.82		.054	.094	.35	.071				
0"	9.4	9' 0"	+135°	52 465	31 985	24.29	62.40	—	.059	.087	.33	.07	—		R, C	
0"	10.4	7' 6"	180°	55 405	31 085	22.75	58.70	—	.057	.095	.30	.07	—		R, C	
6"	9.9	14' 3"	0°	55 685	38 035	25.75	64.60	—	.055	.092	.29	.07	—	Bessemer Steel	C	12" Extra Strong 65.00 Lbs.
0"	8.5	4' 6"	− 20°	53 980	30 805	21.58	51.70	—	.058	.097	.32	.075	—		R, C	
6"	8.9	4' 9"	180°	54 385	31 075	24.13	60.80	—	.055	.092	.32	.075	—		R, C	
0"	9.4			54 384	32 597	23.70	59.68		.057	.093	.31	.072				
0"	12.8	8' 6"	0°	58 005	35 635	28.34	59.40	—	.067	.111	.38	.08	—		R, C	
0"	10.2	5' 6"	0°	59 565	37 245	23.29	57.30	—	.035	.109	.40	.07	—		R, C	
0"	11.1	14' 0"	+ 25°	56 880	34 360	25.63	60.80	—	.055	.098	.37	.075	—	Bessemer Steel	R, C	12½" Casing 13" O.D.
0"	10.2	13' 6"	+ 10°	61 760	41 365	20.17	56.10	—	.048	.102	.34	.08	—		R, C	
0"	12.8	6' 6"	+ 30°	58 225	35 875	25.21	59.60	—	.052	.089	.39	.075	—		R, C	
0"	11.1			58 887	36 896	24.53	58.64		.051	.100	.38	.076				
8	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34

...T OF PRINCIPAL RESULTS OF TESTS, SERIES 2.

SERIES 2 { SHOWING THE INFLUENCE OF OUTSIDE DIAMETER AND THICKNESS OF WALL ON COLLAPSING PRESSURE, for lengths of 20 feet, between end connections tending to hold the tube to a circular form

REMARKS

201. Local depression at weld, 4'-1" from end, about one inch in diameter by .01" or .02" deep.

204. Two splits in weld, each about 6" long

240. Split in weld about 16" long.

249. Collapse commences at coupling

252. Split at weld, 1'-6" from end of tube

253. Split 3'-0" long at weld.

260. Defective pipe.

261. Split 6" long at 10'-3" from end of tube.

262. Tested to 2600 lbs. Pipe held at this pressure for about two minutes, but showed no sign of rupture.

263. Split 6" long, 15'-4" from end, at −5° from weld.

279. Split 12" long, 18'-9" from end, at −45° from weld.

282. Collapsed at the end.

310. Had much trouble getting heads screwed on tight. Collapsed finally on fourth trial. Other times leaks prevented getting sufficient pressure.

311. Gages rose to 1650 lbs then dropped. Pipe was removed from tank and the heads screwed up tighter. When replaced the gages rose to 2250 lbs. and dropped again, owing to a leak in the test head. Finally collapsed as recorded. Collapse begins in the end. The coupling split when head was unscrewed, and threads were badly twisted.

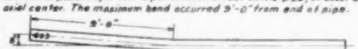
313. Rupture 2'-0" long.

320. Split in weld

324. Split in weld.

400. This test is doubtful. At 400 lbs. the gages began to fall. Pumping speed was then increased slightly but gages continued to fall and pump was stopped. While pressure was on, constant watch had been kept for leaks in the pressure system. No leak was discovered. Writer distinguished no sound as of collapse, nor did the two workmen standing nearest the tank. Two other men claimed to have heard a faint sound but did not associate it with collapse. When the pump was started again, the gages responded quickly, rising at about one pound per second to 150 lbs. at which point water issued from the vent. Specimen was withdrawn completely collapsed.

483. This specimen was distorted under about 90,000 lbs. compressive stress, when pressing heads on. Though pressure was removed as soon as possible, it left a permanent offset in the pipe, of about 8" from the axial center. The maximum bend occurred 9'-0" from end of pipe.



Pipe was tested without being straightened.

REMARKS

426. This pipe plainly showed evidence of "red-shortness". Marking was noticeable at 11'-0", 180° from weld, and less so at other points.

440. Collapsing strained 20 foot end away from its support, i.e. away from press fitted head.

444. Complete rupture at weld between 17'-0" and 19'-0". Collapse distortion at 20 feet end pulled tube walls away from press fitted head.

445. Collapsed gradually without report. Pressure withdrawn as soon as tangible fall of gages was noticed.

446. See note 445. This specimen collapsed in two places. Between 1'-0" and 5'-0" the distortion was scarcely noticeable to the eye, though discernable. Between 9'-0" and 20'-0" it was more marked. Both collapsed regions were determined by calipering. Axes of collapse for both regions were approximately in the same plane.

447. See note 445.

448. See note 445.

449. See note 445.

450. Collapsed without report.

451. Specimen badly pitted throughout length. See note 450.

453. See note 451.

454. See note 450.

460. Dinged at 10'-11", 11'-11½", 12'-2", 12'-10", 13'-3", and 14'-0". Offset from axial center about 1", maximum bend occurring at about 12'-0".

461. Owing to unsteadiness of pump, collapsing pressure could not be accurately read.

481. Pressure as observed and recorded is not correct. After collapse it was discovered that the ¾" valve between the tank and the gages had been nearly closed against the pressure.

491. Approximate pressure recorded. Precise reading not observed owing to sudden acceleration of pump after stalling at 2350 lbs.

505. Collapsed region bound within I.D. of testing cylinder. Specimen removed by force.

506. See note 505.

508. See note 505.

509. See note 508.

510. Collapsed without report. Distortion slight. Difference between major and minor axes approximates five-eighths of an inch. Pressure continued a few strokes after gages fell.

511. Collapsed without report. Distortion slight. Difference between major and minor axes approximates one-half an inch. The pump was stopped as soon as gages fell.

REMARK

512. See note 511.

513. See note 511.

514. See note 511.

The 1000 lb. Gage is mark
" 3000 lb " " "
" 8500 lb " " "

Readings on Gages B and D a
on Gage C

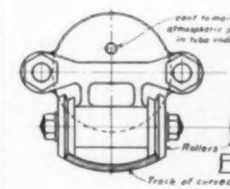


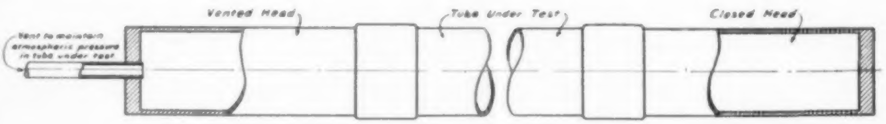
FIG. 42.—TABULAR STATEMENT OF

PRINCIPAL RESULTS OF COLLAPSING TESTS ON NATIONAL TUBE CO'S. LAP-WELDED BESSEMER STEEL TUBES

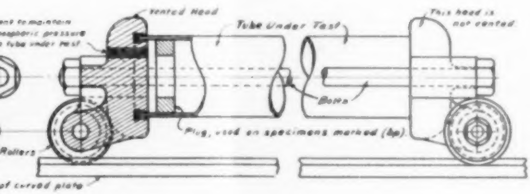
CONDUCTED BY PROF. R.T. STEWART, 1902-4. F.P.K. 1906.

MARKS

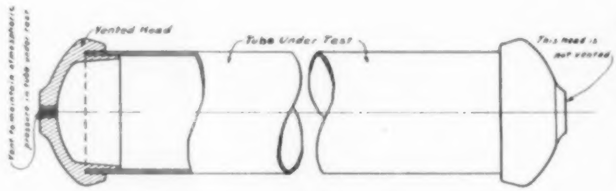
is marked B
" " C
" " D
and D are reduced to Readings



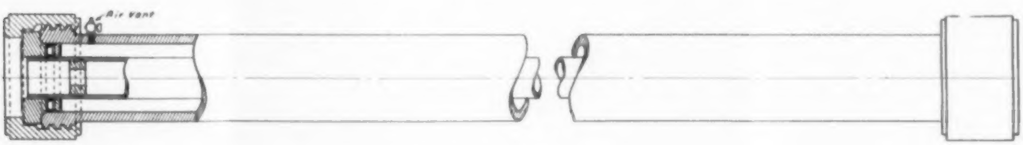
Coupled Test Head (a).



Bolted Test Head (b).



Press Fitted Test Head (c).



Small Test Cylinder (d).

NT OF PRINCIPAL RESULTS OF TESTS, SERIES 2.

COLLAPSING PRESSURES.— Abstract from Log of Tests Conducted by Prof. R. T. Stewart, 1902-04, on National Tube Co's Lap-welded Bessemer Steel Tubes, 20 Foot Lengths, to which is added a Comparison with Calculated Values, by Formulae A&B. Made by E. E. S. under direction of R. T. S., 1906.

Tests Grouped according to Outside Diameter and Arranged in Order of Thicknesses of Tubes.

Number of Test	Outside Diameter, Inches		Thickness of Wall, Inches		Actual Plain End Weight, Lbs per Ft	Collapsing Pressure Pounds per Square Inch			Commercial Designation of Tube, as Reported.
	Nominal	Actual	Nominal	Computed from Wgt		Observed	Calc'd by Formula	% Variation from Calc'd	
480		2.999	0.109	0.110	3.40	1550	1793	−13.6	3" Standard Boiler Tubing, 3.33*
482		3.006	0.109	0.110	3.40	1630	1786	− 8.7	" " " " "
483		3.001	0.109	0.110	3.40	1725	1791	− 3.7	" " " " "
484		2.993	0.109	0.111	3.43	2025	1828	+10.8	" " " " "
487	3.000	2.998	0.120	0.112	3.45	1960	1852	+ 5.8	" Locomotive " " 3.63*
485		2.986	0.120	0.112	3.45	1800	1865	− 3.5	" " " " "
486		2.997	0.120	0.112	3.45	1850	1853	− 0.2	" " " " "
489		2.992	0.120	0.113	3.48	2175	1887	+15.3	" " " " "
488		2.998	0.120	0.114	3.53	2025	1910	+ 6.0	" " " " "
Average		2.997		0.112	3.44	1860	1841	+ 0.9	
491	3.000	2.987	0.13	0.139	4.23	2575	2647	− 2.7	3" Special, 4.57 lbs.
490		2.997		0.147	4.48	3350	2865	+16.9	
Average		2.992		0.143	4.35	2962	2756	+ 7.1	
498		3.000		0.182	5.48	3700	3872	− 4.4	3" Special, 5.42 lbs.
499		2.994		0.187	5.65	4200	4085	+ 2.8	
495	3.000	2.990	0.18	0.190	5.68	4200	4122	+ 1.9	
497		2.997		0.190	5.68	4175	4109	+ 1.6	
496		2.996		0.191	5.73	4200	4140	+ 1.5	
Average		2.995		0.188	5.64	4095	4066	+ 0.7	
464		3.992		0.114	4.71	860	1089	−21.0	4" Converse Joint, 4.89 lbs.
460		3.990		0.119	4.91	925	1199	−22.8	
463	4.000	4.001	0.120	0.120	4.98	1030	1214	−15.2	
461		3.990		0.122	5.06	975	1264	−22.9	
462		3.990		0.122	5.03	1030	1264	−18.5	
Average		3.993		0.119	4.94	964	1206	−20.1	
469		4.017		0.169	6.94	2160	2261	− 4.5	3½" English, 7.35 lbs.
465		4.010		0.173	7.08	2050	2353	−12.9	
467	4.000	4.012	0.18	0.173	7.11	2425	2351	+ 3.1	
466		4.014		0.178	7.28	2225	2457	− 9.4	
468		4.018		0.184	7.53	2540	2583	− 1.7	
Average		4.014		0.175	7.19	2280	2401	− 5.1	
474		4.019		0.205	8.32	3075	3035	+ 1.3	3½" Full Weight, 9.00 lbs.
472		4.029		0.210	8.57	3150	3132	+ 0.6	
471	4.000	4.024	0.226	0.213	8.67	3125	3202	− 2.4	
470		4.029		0.216	8.77	3125	3239	− 3.5	
473		4.028		0.217	8.82	3375	3283	+ 2.8	
Average		4.026		0.212	8.63	3170	3178	− 0.2	
475		4.020		0.324	12.79	5525	5600	− 1.3	3½" Extra Strong, 12.47 lbs.
478		4.011		0.326	12.85	5600	5659	− 1.0	
477	4.000	4.016	0.321	0.326	12.83	5625	5650	− 0.4	
479		4.010		0.328	12.89	5425	5704	− 4.9	
476		4.012		0.332	13.05	5625	5786	− 2.8	
Average		4.014		0.327	12.89	5560	5680	− 2.1	
200		6.018		0.123	7.77	450	424	+ 6.1	6" Converse Joint, 8.26 lbs.
201		6.026		0.127	8.00	500	462	+ 8.2	
209		6.016		0.128	8.07	485	475	+ 2.1	
206		6.021		0.129	8.12	480	485	− 1.0	
203	6.000	6.019	0.134	0.130	8.19	540	496	+ 8.9	
202		6.015		0.130	8.19	575	498	+15.5	
204		6.010		0.131	8.24	530	511	+ 3.7	
205		6.023		0.131	8.22	530	507	+ 4.5	
207		6.013		0.134	8.45	640	547	+17.0	
208		6.013		0.135	8.45	510	560	− 8.9	
Average		6.017		0.130	8.17	524	497	+ 5.6	
1	2	3	4	5	6	7	8	9	10

FIG. 43.

COLLAPSING PRESSURES.— Abstract from Log of Tests Conducted by Prof. R. T. Stewart, 1902-04, on National Tube Co.'s Lap-welded Bessemer Steel Tubes, 20 Foot Lengths, to which is added a Comparison with Calculated Values, by Formulae A & B. Made by E. E. S. under direction of R. T. S., 1905.

Tests Grouped according to Outside Diameter and Arranged in Order of Thicknesses of Tubes.

Number of Test	Outside Diameter, Inches		Thickness of Wall Inches		Actual Plain End Weight, Lbs. per Ft.	Collapsing Pressure Pounds per Square Inch			Commercial Designation of Tube, as Reported.
	Nominal	Actual	Nominal	Computed from Wgt.		Observed	Calc'd by Formula	% Variation from Calc'd	
216		6.028	0.156	0.155	9.75	760	843	- 9.8	5⅝" Casing, 10.46 lbs.
222		6.037	0.156	0.156	9.79	880	854	+ 3.1	" " " "
213		6.037	0.156	0.164	10.30	850	968	-12.2	" " " "
224		6.026	0.156	0.166	10.40	1110	1002	+10.8	" " " "
225		6.038	0.156	0.166	10.39	715	997	-28.3	" " " "
221		6.034	0.156	0.166	10.40	760	998	-23.8	" " " "
212		6.011	0.156	0.166	10.35	950	1008	- 5.8	" " " "
219		6.021	0.156	0.166	10.39	980	1003	- 2.3	" " " "
210	6.000	6.007	0.156	0.167	10.44	1025	1024	+ 0.1	" " " "
238		6.037	0.156	0.167	10.45	775	1011	-23.3	" " " "
291		6.034	0.156	0.168	10.50	1050	1027	+ 2.2	" " " "
218		6.008	0.156	0.168	10.50	860	1037	-17.1	" " " "
215		6.033	0.156	0.168	10.55	1010	1028	- 1.7	" " " "
211		6.024	0.156	0.169	10.59	930	1046	-11.1	" " " "
223		6.026	0.156	0.169	10.54	1070	1045	+ 2.4	" " " "
243		6.034	0.220	0.170	10.62	875	1056	-17.1	" " 14.20 "
214		6.031	0.156	0.171	10.69	1090	1071	+ 1.8	" " 10.46 "
217		6.038	0.156	0.175	10.92	1010	1126	-10.3	" " " "
Average		6.028		0.167	10.42	928	1008	- 7.9	
226		6.028	0.156	0.178	11.14	1425	1173	+21.5	5⅝" Casing, 10.46 lbs.
247		6.055	0.220	0.179	11.22	1010	1176	-14.1	" " 14.20 "
240		6.035	0.220	0.179	11.19	1095	1185	- 7.6	" " " "
259		6.024	0.203	0.182	11.35	1100	1233	-10.8	" " 12.04 "
258		6.013	0.203	0.182	11.30	1350	1237	+ 9.1	" " " "
253		6.020	0.203	0.184	11.45	950	1263	-24.8	" " " "
242	6.000	6.035	0.220	0.185	11.57	1272	1271	+ 0.1	" " 14.20 "
256		6.021	0.203	0.185	11.55	1450	1277	+13.5	" " 12.04 "
257		6.025	0.203	0.189	11.77	1250	1333	- 6.2	" " " "
255		6.028	0.203	0.192	11.98	790	1375	-42.5	" " " "
250		6.016	0.203	0.192	11.96	1450	1380	+ 5.1	" " " "
245		6.010	0.220	0.193	11.97	1375	1397	- 1.6	" " 14.20 "
244		5.998	0.220	0.193	11.97	1750	1403	+24.7	" " " "
Average		6.024		0.186	11.57	1251	1285	- 2.6	
251		6.016	0.203	0.214	13.25	1750	1697	+ 3.1	5⅝" Casing, 12.04 lbs.
241		6.021	0.220	0.219	13.57	1600	1767	- 9.4	" " 14.20 "
254		6.005	0.203	0.220	13.59	1550	1789	-13.4	" " 12.04 "
252	6.000	6.020	0.203	0.222	13.73	2075	1810	+14.6	" " " "
*249		6.036	0.220	0.228	14.15	1200	1888	-36.4	" " 14.20 "
246		6.001	0.220	0.230	14.17	1900	1936	- 1.9	" " " "
248		6.004	0.220	0.232	14.30	1800	1963	- 8.3	" " " "
Average		6.011		0.222	13.77	1779	1827	- 2.5	*Defective, not in averages.
413		6.072	0.271	0.250	15.33	2220	2183	+ 1.7	5⅝" Casing, 16.70 lbs.
261		6.027	0.271	0.251	15.49	1750	2223	-21.3	" " " "
279		6.072	0.271	0.254	15.79	2360	2240	+ 5.4	" " " "
276		6.033	0.271	0.259	15.94	2475	2335	+ 6.0	" " " "
*260		6.054	0.271	0.260	16.06	1755	2336	-24.9	" " " "
264		6.022	0.271	0.260	15.99	2450	2356	+ 4.0	" " " "
416		6.042	0.271	0.262	16.16	2555	2372	+ 7.7	" " " "
†262		6.033	0.271	0.263	16.20	2600†	2392	+ 8.7	" " " "
282		6.024	0.271	0.267	16.44	2250	2456	- 8.4	" " " "
442	6.000	5.990	0.28	0.269	16.45	2590	2506	+ 3.4	6" O.D. Special, 17.12 lbs.
441		5.993	0.28	0.269	16.45	2150	2504	-14.1	" " " "
273		6.021	0.271	0.270	16.59	2270	2501	- 9.2	5⅝" Casing, 16.70 lbs.
283		6.022	0.271	0.270	16.55	2550	2500	+ 2.0	" " " "
443		5.998	0.28	0.271	16.55	2460	2530	- 2.8	6" O.D. Special, 17.12 lbs.
417		6.042	0.271	0.271	16.71	2400	2501	- 4.0	5⅝" Casing, 16.70 lbs.
419		6.044	0.271	0.271	16.73	2575	2500	+ 3.0	" " " "
440		5.991	0.28	0.273	16.68	2780	2563	+ 8.5	6" O.D. Special, 17.12 lbs.
444		5.993	0.28	0.274	16.73	2455	2576	- 4.7	" " " "
418		6.044	0.271	0.277	17.08	2890	2586	+11.8	5⅝" Casing, 16.70 lbs.
263		6.025	0.271	0.280	17.17	2600	2642	- 1.6	" " " "
Average		6.026		0.266	16.38	2441	2446	- 0.2	*Defective, not in averages. †Did not collapse.
1	2	3	4	5	6	7	8	9	10

FIG. 44.

COLLAPSING PRESSURES.— Abstract from Log of Tests Conducted by Prof R. T. Stewart, 1902-04, on National Tube Co's Lap-welded Bessemer Steel Tubes, 20 Foot Lengths, to which is added a Comparison with Calculated Values, by Formulae A&B. Made by E. E. S. under direction of R. T. S., 1905.

Tests Grouped according to Outside Diameter and Arranged in Order of Thicknesses of Tubes.

Number of Test.	Outside Diameter Inches.		Thickness of Wall Inches.		Actual Plain End Weight, Lbs. per Ft.	Collapsing Pressure Pounds per Square Inch			Commercial Designation of Tube, as Reported.
	Nominal	Actual	Nominal	Computed from Wgt.		Observed	Calc'd by Formula	% Variation from Calc'd	
402	6.625	6.665	0.15	0.152	10.57	585	591	- 1.0	6⅝" O.D. Special, 10.39 lbs.
404		6.657	0.15	0.153	10.59	520	606	-14.2	" " " " "
403		6.661	0.15	0.154	10.67	560	618	- 9.4	" " " " "
*400		6.663	0.15	0.154	10.68	400	617	-35.2	" " " " "
300		6.657	0.172	0.157	10.87	710	658	- 7.9	6¼" Casing, 11.38 lbs.
401		6.658	0.15	0.157	10.87	585	658	-11.1	6⅝" O.D. Special, 10.39 lbs.
Average		6.660		0.155	10.71	592	626	- 8.7	*Defective, not in averages. See log.
303	6.625	6.653	0.172	0.164	11.35	730	750	- 2.7	6¼" Casing, 11.38 lbs.
304		6.652		0.165	11.42	600	764	-21.5	
301		6.657		0.165	11.47	720	762	- 5.5	
302		6.653		0.168	11.62	630	803	-21.5	
Average		6.654		0.166	11.47	670	770	-12.8	
305	6.625	6.680	0.203	0.196	13.57	1100	1157	- 4.9	6¼" Casing, 13.32 lbs.
309		6.676		0.196	13.57	1075	1159	- 7.2	
306		6.693		0.200	13.85	1205	1204	+ 0.1	
307		6.682		0.202	13.99	1275	1234	+ 3.3	
308		6.687		0.205	14.17	1265	1271	- 0.5	
Average		6.684		0.200	13.83	1184	1205	- 1.8	
410	6.625	6.686	0.238	0.241	16.59	1680	1738	- 3.3	6¼" Casing, 17.02 lbs.
413		6.679		0.248	16.99	1710	1832	- 6.7	
414		6.682		0.248	17.01	1615	1831	-11.8	
*313		6.657		0.250	17.09	1820	1869	- 2.6	
314		6.671		0.251	17.20	2175	1875	+16.0	
411		6.684		0.251	17.24	1940	1869	+ 3.8	
312		6.669		0.252	17.25	1975	1889	+ 4.6	
311		6.674		0.254	17.39	2160	1913	+12.9	
412		6.675		0.256	17.51	1780	1938	- 8.2	
Average		6.677		0.250	17.15	1879	1861	+ 0.9	*Iron, not in averages.
310	6.625	6.661	0.238	0.260	17.74	2275	1997	+13.9	6¼" Casing, 17.02 lbs.
408		6.651	0.280	0.261	17.81	2060	2015	+ 2.2	6" Full Weight, 18.76 lbs.
405		6.650	0.280	0.262	17.85	2136	2028	+ 5.3	" " " " "
406		6.653	0.280	0.262	17.85	1975	2027	- 2.6	" " " " "
407		6.655	0.280	0.278	18.94	2560	2239	+14.6	" " " " "
409		6.659	0.280	0.284	19.35	2340	2313	+ 1.2	" " " " "
Average		6.652		0.268	18.26	2224	2102	+ 5.8	
439	7.000	7.009	0.148	0.153	11.21	515	513	+ 0.4	7" Converse Joint, 10.65 lbs.
317		7.043	0.180	0.184	11.31	575	515	+11.6	6⅝" Casing, 12.34 lbs.
437		7.009	0.148	0.155	11.31	605	533	+13.5	7" Converse Joint, 10.65 lbs.
316		7.040	0.180	0.158	11.58	550	559	- 1.6	6⅝" Casing, 12.34 lbs.
315		7.046	0.180	0.158	11.61	570	558	+ 2.2	" " " "
435		7.008	0.148	0.161	11.75	665	605	+ 9.9	7" Converse Joint, 10.65 lbs.
438		7.015	0.148	0.162	11.85	675	615	+ 9.8	" " " " "
319		7.059	0.180	0.163	11.96	540	615	-12.2	6⅝" Casing, 12.34 lbs.
318		7.033	0.180	0.166	12.13	580	660	-12.1	" " " "
436		7.015	0.148	0.168	12.25	645	690	- 6.5	7" Converse Joint, 10.65 lbs.
Average		7.028		0.160	11.70	592	586	+ 1.5	
322	7.000	7.057	0.248	0.233	16.97	1525	1476	+ 3.3	6⅝" Casing, 17.51 lbs.
321		7.056		0.241	17.50	1575	1577	- 0.1	
320		7.050		0.245	17.76	1775	1626	+ 9.2	
323		7.045		0.245	17.77	1675	1628	+ 2.9	
324		7.047		0.248	18.02	1850	1664	+11.2	
Average		7.050		0.242	17.60	1680	1594	+ 5.3	
432	7.000	7.018	0.28	0.268	19.33	1835	1924	- 4.6	7" O.D. Special, 20.12 lbs.
434		6.984		0.269	19.34	1975	1952	+ 1.2	
431		6.975		0.283	20.23	2300	2130	+ 8.0	
433		6.979		0.287	20.47	2180	2181	- 0.0	
430		6.984		0.290	20.74	2445	2213	+10.5	
Average		6.987		0.279	20.02	2147	2080	+ 3.0	
1	2	3	4	5	6	7	8	9	10

FIG. 45.

COLLAPSING PRESSURES.— Abstract from Log of Tests Conducted by Prof R. T. Stewart, 1902-04, on National Tube Co's Lap-welded Bessemer Steel Tubes, 20 Foot Lengths, to which is added a Comparison with Calculated Values, by Formulae A&B. Made by E. E. S. under direction of R. T. S., 1905.

Tests Grouped according to Outside Diameter and Arranged in Order of Thicknesses of Tubes.

Number of Test.	Outside Diameter, Inches. Nominal	Outside Diameter, Inches. Actual.	Thickness of Wall, Inches. Nominal	Thickness of Wall, Inches. Computed from Wgt.	Actual Plain End Weight, Lbs. per Ft.	Collapsing Pressure, Pounds per Square Inch. Observed	Calc'd by Formula	% Variation from Calc'd	Commercial Designation of Tube, as Reported.
1		8.657		0.176	15.92	450	418	+ 7.7	
4		8.640		0.183	16.54	450	469	- 4.0	
3	8.625	8.641	0.180	0.186	16.77	535	491	+ 9.0	8⅝" Casing, 16.07 lbs.
2		8.637		0.191	17.24	625	534	+17.0	
5		8.638		0.191	17.23	620	533	+16.3	
Average		8.643		0.185	16.74	536	489	+ 9.2	
26	8.625	8.604	0.229	0.219	19.57	870	820	+ 6.1	8⅝" Casing, 20.10 lbs.
52		8.664	0.271	0.258	23.13	1320	1195	+10.5	8⅝" Casing, 24.38 lbs.
54		8.660	0.271	0.262	23.52	1485	1236	+20.1	" " " "
99		8.663	0.322	0.264	23.66	1375	1255	+ 9.6	8" Line Pipe, 28.18 lbs.
50		8.660	0.271	0.271	24.28	1435	1326	+ 8.2	8⅝" Casing, 24.38 lbs.
53	8.625	8.660	0.271	0.272	24.32	1520	1336	+13.8	" " " "
76		8.663	0.281	0.272	24.38	1410	1335	+ 5.6	8" Line Pipe, 25.00 lbs.
75		8.640	0.281	0.274	24.44	1375	1363	+ 0.9	" " " " "
51		8.668	0.271	0.274	24.54	1430	1354	+ 5.6	8⅝" Casing, 24.38 lbs.
*77		8.660	0.281	0.274	24.47	1275	1356	- 6.0	8" Line Pipe, 25.00 lbs.
*78		8.660	0.281	0.280	25.08	1250	1416	-11.7	" " " " "
Average		8.660		0.268	24.03	1419	1300	+ 9.3	*Iron, not in averages.
101		8.647	0.322	0.294	26.27	1650	1561	+ 5.7	8" Line Pipe, 28.18 lbs.
100		8.646	0.322	0.297	26.44	1710	1591	+ 7.5	" " " " "
424		8.671	0.322	0.302	26.98	1575	1633	- 3.5	8" Full Weight, 28.18 lbs.
102		8.656	0.322	0.303	27.00	1960	1648	+18.9	8" Line Pipe, 28.18 lbs.
423	8.625	8.649	0.322	0.303	27.00	1830	1650	+10.9	8" Full Weight, 28.18 lbs.
421		8.666	0.322	0.303	27.09	1935	1644	+17.7	" " " " "
422		8.672	0.322	0.307	27.39	1635	1682	- 2.8	" " " " "
†98		8.666	0.322	0.308	27.44	1735	1694	+ 2.4	8" Line Pipe, 28.18 lbs.
420		8.658	0.322	0.310	27.64	1805	1717	+ 5.1	8" Full Weight, 28.18 lbs.
Average		8.658		0.302	26.98	1762	1641	+ 7.4	†O. H. Steel, not in averages.
*427		8.678		0.346	30.82	1830	2070	-11.6	
*425		8.666		0.349	31.00	2180	2104	+ 3.6	
*428	8.625	8.664	0.363	0.353	31.36	2045	2145	- 4.7	8" Oil Well Tubing, 32.00 lbs.
*429		8.683		0.356	31.62	1930	2167	-10.9	
*426		8.672		0.364	32.30	2155	2252	- 4.3	
Average		8.673		0.354	31.42	2028	2148	- 5.6	*Iron, group rejected.
449		10.055		0.157	16.55	210	219	- 4.1	
447		10.045		0.166	17.50	240	250	- 4.0	
445	10.000	10.037	0.156	0.167	17.58	210	254	-17.3	10" Converse Joint, 16.18 lbs.
446		10.031		0.167	17.59	225	254	-11.4	
448		10.035		0.170	17.99	240	265	- 9.4	
Average		10.041		0.165	17.43	225	248	- 9.2	
452		10.005		0.185	19.43	305	327	- 6.7	
453		10.033		0.190	19.94	395	347	+13.8	
451	10.000	10.029	0.203	0.194	20.35	390	367	+ 6.3	10" Boiler Tubing, 21.00 lbs.
454		10.037		0.195	20.54	400	371	+ 7.8	
450		10.027		0.206	21.57	425	430	- 1.2	
Average		10.026		0.194	20.37	383	368	+ 4.0	
456		9.990		0.312	32.27	1350	1321	+ 2.2	
459		10.023		0.314	32.61	1385	1329	+ 4.2	
457	10.000	9.989	0.30	0.317	32.71	1275	1364	- 6.5	10" O.D. Special, 31.07 lbs.
458		10.003		0.317	32.81	1305	1360	- 4.0	
455		10.000		0.319	33.01	1280	1379	- 7.2	
Average		10.001		0.316	32.68	1319	1351	- 2.3	
1	2	3	4	5		7	8	9	10

Formula A, $P = 1000\left(1-\sqrt{1-1600\frac{t^2}{d^2}}\right)$. Formula B, $P = 86670\frac{t}{d} - 1386$.

P = collapsing fluid pressure, in lbs per sq in., t = thickness of wall in inches, d = outside dia of tube in inches. Formula B applies only to values of $\frac{t}{d}$ greater than 0.227, or pressures greater than 581 pounds, while formula A applies only to values less than these.

FIG. 46.

After a fruitless effort to derive a satisfactory formula on the basis of three variables, which, when plotted, would, of course, be a surface in space, the thought happily presented itself that two of these variables, t and d, could be replaced by their quotient, or $\frac{t}{d}$ which, of course, might be treated as a single variable. By the adoption of this expedient, matters were greatly simplified, since it thus became possible to plot the results of the tests for all diameters and thicknesses of wall on a plane surface.

Fig. 47 shows the group averages of Figs. 43-46 plotted in this manner, that is to say, to a vertical scale representing fluid-collapsing pressures in pounds per square inch, and a horizontal scale representing the quotient arising from dividing the thickness of wall by the outside diameter, both being expressed in inches.

Formula B.—By an inspection of Fig. 47 it became apparent that the bulk of the group averages could be represented by a straight-line formula, indeed all of them could be thus represented with the exception of the few having values of thickness divided by outside diameter less than 0.023. In other words, about 93 per cent. of the group averages of Figs. 43-46 can be thus represented.

On this basis then, for values of $\frac{t}{d}$ greater than 0.023, formula B was deduced, it being as follows:

$$P = 86{,}670 \frac{t}{d} - 1386 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

Where P = collapsing pressure, lbs. per sq. inch,
d = outside diameter of tube in inches,
t = thickness of wall in inches.

Remembering that this same formula might also have been arrived at by the substitution of proper empirical constants in a similar formula for a theoretically perfect tube, and further, since Fig. 47 shows no apparent deviation from straightness on the upward course, it was not thought necessary to set an upper limit to the value of $\frac{t}{d}$ in the application of this formula, believing that it will give substantially correct results for all commercial lap-welded Bessemer-steel tubes whose thickness divided by the outside diameter is greater than 0.023.

Formula A.—This formula for values of $\frac{t}{d}$ less than 0.023

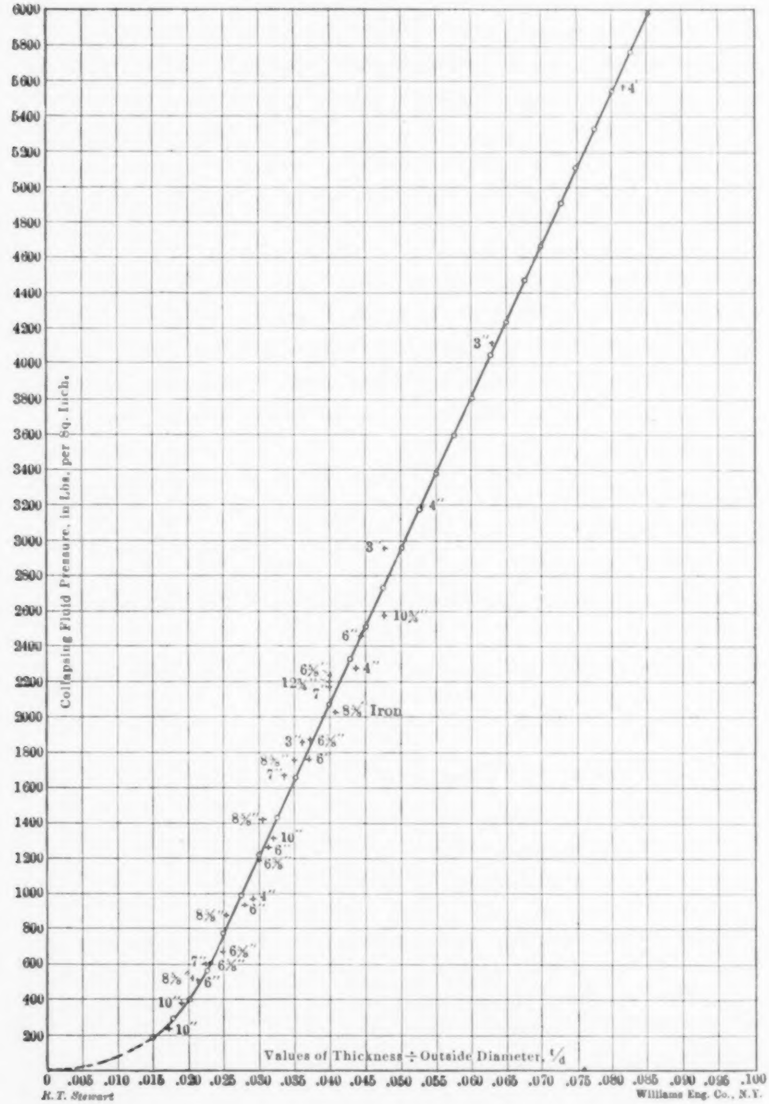


FIG. 47.—CHART SHOWING ACTUAL AND CALCULATED COLLAPSING PRESSURES OF NATIONAL TUBE CO.'S BESSEMER STEEL LAP-WELDED TUBES PLOTTED TO THICKNESS ÷ OUTSIDE DIAMETER, OR $t/_d$. BASED ON TESTS BY PROF. STEWART ON 20-FOOT LENGTHS.

Note that group averages are represented by crosses (+), the attached figures indicating outside diameter, while values calculated by means of formulæ A and B are represented by circles (○).

was derived upon the assumption that when plotted upon Fig. 47 the resulting curve should be tangent to the straight line representing formula B, and be also tangent to the horizontal axis at the origin O. This arbitrary assumption gave a formula which represented very satisfactorily the few experiments in which $\frac{t}{d}$ was less than 0.023. The formula thus obtained is

$$P = 1000\left(1 - \sqrt{1 - 1600\frac{t^2}{d^2}}\right) \quad . \quad . \quad . \quad . \quad \text{(A)}$$

Where P, d and t are the same as for formula B.

This formula should be used for values of $\frac{t}{d}$ less than 0.023 and for P greater than $\frac{1}{22}\sqrt{d}$.

Since constructing the charts and tables contained in this paper, it was discovered that a formula having a rational form with empirical constants could be substituted for the purely empirical formula A. This formula, in addition to involving theoretical considerations of elasticity, is much the simpler of the two. It is applicable only to tubes having relatively thin walls, that is to say, to those having values of $\frac{t}{d}$ less than 0.023, and is

$$P = 50{,}210{,}000\left(\frac{t}{d}\right)^3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(G)}$$

Where P, d and t are the same as for formula A.

Either formula A or G represents satisfactorily the results of the experiments made on thin-walled tubes, that is, those in which $\frac{t}{d}$ is less than 0.023, but probably formula G will permit of the greater exterpolation.

The following formulæ are meant for application in case the outside diameter and plain-end weight are given. They were derived from formulæ A and B and are

$$P = 1000\left(1 - \sqrt{149.8\frac{w}{d^2} - 799 + 800\sqrt{1 - 0.375\frac{w}{d^2}}}\right) . \text{(C)}$$

$$P = 41{,}950 - 26{,}520\sqrt{2.67 - \frac{w}{d^2}} \quad . \quad . \quad . \quad . \quad \text{(D)}$$

Where w = the plain-end weight of tube in pounds per foot, while P and d are the same as for formulæ A and B.

Formula C is for values of $\frac{w}{d^2}$ less than 0.237 and P less than 581 lbs., while formula D is for values greater than these.

Charts of Actual and Calculated Collapsing Pressures.—Figs. 48-50 show a comparison of the results obtained by actual test with the corresponding calculated values, plotted to a vertical scale representing collapsing pressures in pounds per square inch, and a horizontal scale representing the thickness of wall in decimals of an inch. It will be noted that Fig. 48 is for the experimental tubes having outside diameters of 3, 6 and 10 inches, the diameter being written in each case on the margin at the right-hand end of the line representing the tube. Similarly, Figs. 49 and 50 were constructed for the tubes having outside diameters of respectively 4 and 7 inches, and 6⅝ and 8⅝ inches.

The lines on these Charts were plotted from values calculated by means of formulæ A and B, representing the most probable values for the collapsing pressures of lap-welded Bessemer-steel tubes in lengths of 20 feet between transverse joints tending to hold the tube to a circular form. The center of each small circle lying on these lines represents a plotted calculated value.

The actual collapsing pressures of the different tubes tested, plotted to the same scales as the calculated values, are represented by crosses (+) for those having outside diameters of 3, 10, 4, 7 and 8⅝ inch, and by tees (T) for the 6 and the 6⅝ inch. In a number of instances, in order to avoid confusion, the characters being very close together, it became necessary to omit a part of the cross (+), in which case it became a tee (T), and likewise a part of the tee, it thus appearing as an angle or ell (L).

Group averages are represented on these charts by means of combined crosses and circles (-o-).

It will be observed that the group averages of the actual collapsing pressures lie very close to the corresponding values calculated by means of formulæ A and B. For the actual variation in per cent., see Figs. 43-46, column 9, which gives the variation for the individual tests as well as for the group averages.

Formulæ A and B being based upon the results of all the experiments on the 20-foot lengths of the lap-welded Bessemer-steel tubes tested, excepting the three that proved to be defective, it is clear that the curves plotted on these charts represent average values for the extreme range in thickness of wall for each of the seven diameters tested.

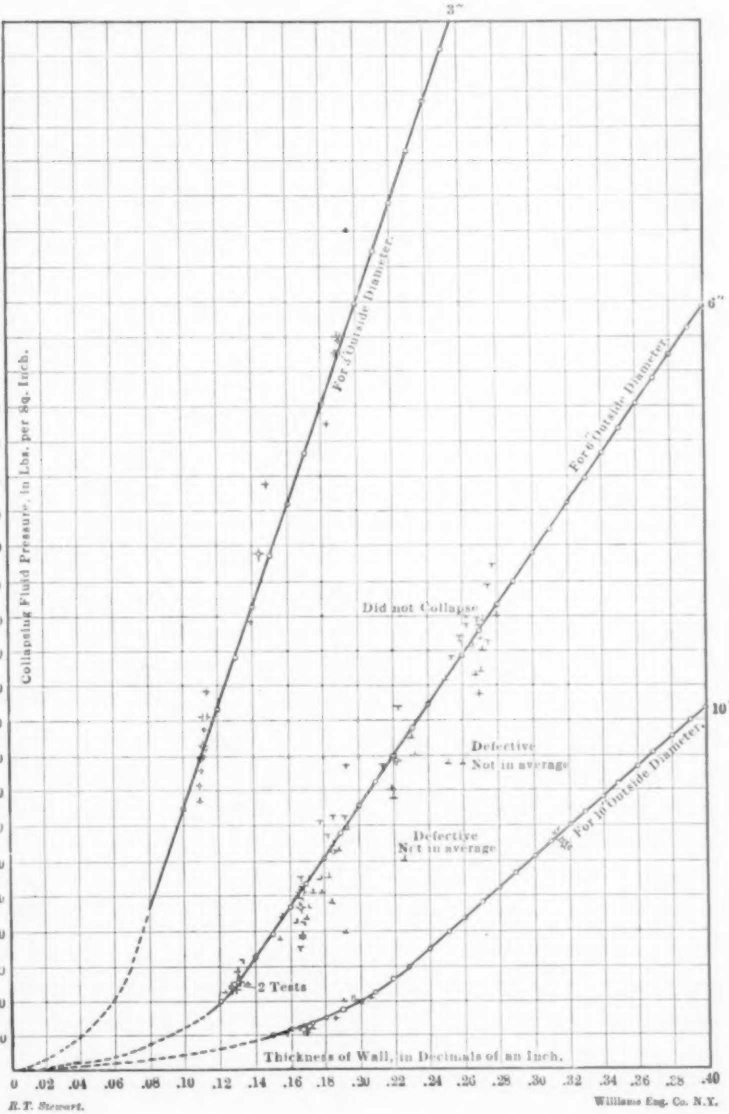


FIG. 48.—CHART SHOWING ACTUAL AND CALCULATED COLLAPSING PRESSURES OF NATIONAL TUBE CO.'S BESSEMER-STEEL LAP-WELDED TUBES PLOTTED TO THICKNESS OF WALL. FOR OUTSIDE DIAMETERS OF 3, 6 AND 10 INCHES, IN 20-FOOT LENGTHS. BASED ON TESTS BY PROF. STEWART, 1902-4.

Note that individual experiments are represented by crosses (+), calculated values by circles (○), and group averages by combined circles and crosse (-ǫ-)

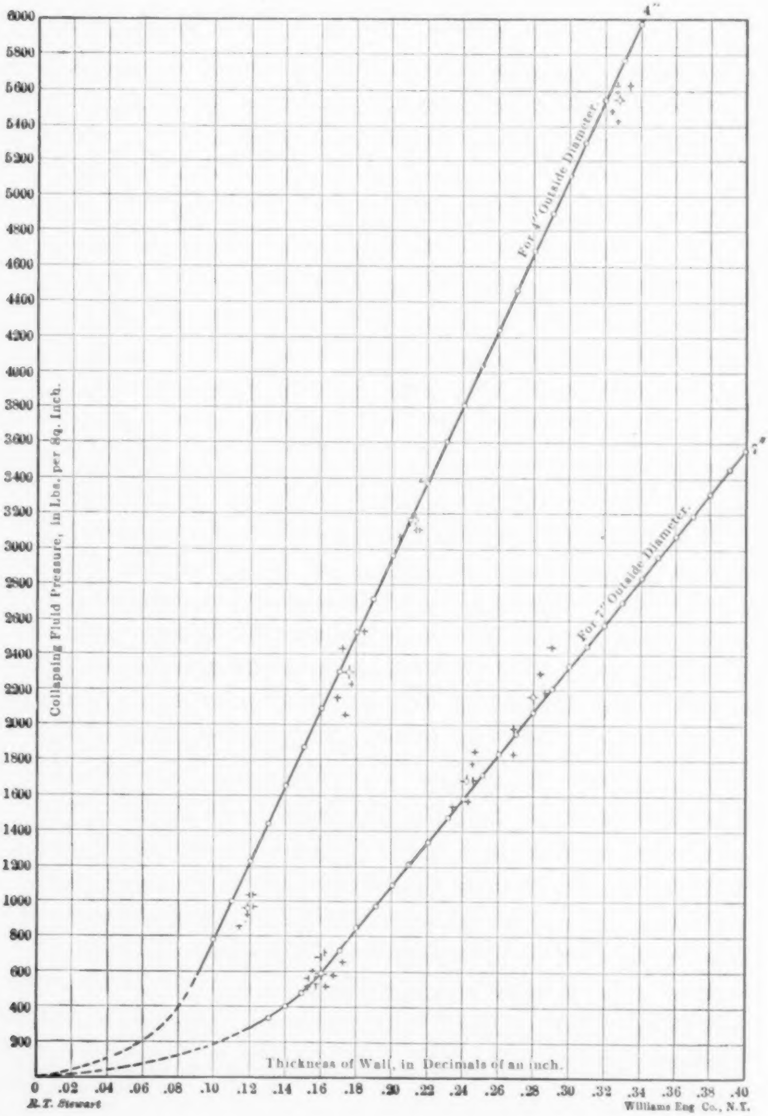


FIG. 49.—CHART SHOWING ACTUAL AND CALCULATED COLLAPSING PRESSURES OF NATIONAL TUBE CO.'S BESSEMER-STEEL LAP-WELDED TUBES PLOTTED TO THICKNESS OF WALL. FOR OUTSIDE DIAMETERS OF 4 AND 7 INCHES, IN 20-FOOT LENGTHS. BASED ON TESTS BY PROF. STEWART, 1902-4.

Note that individual experiments are represented by crosses (+) calculated values by circles (○) and group averages by combined circles and crosses(-○-).

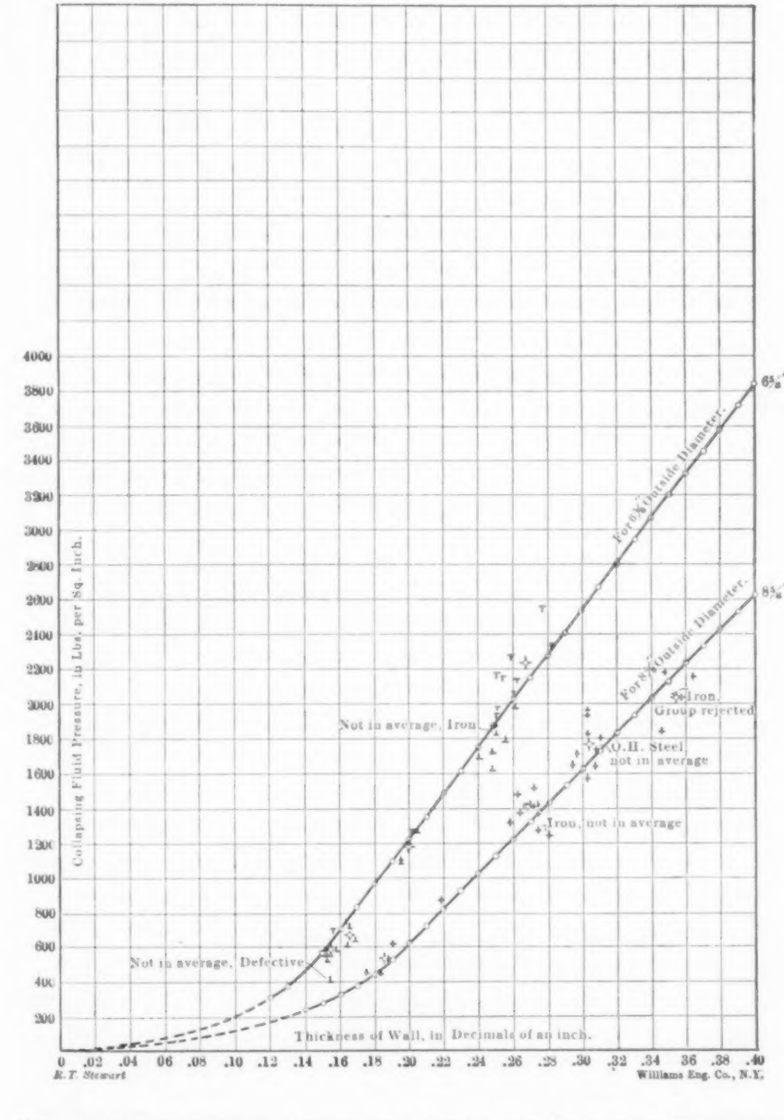


FIG. 50.—CHART SHOWING ACTUAL AND CALCULATED COLLAPSING PRESSURES OF NATIONAL TUBE CO.'S BESSEMER-STEEL LAP-WELDED TUBES PLOTTED TO THICKNESS OF WALL. FOR OUTSIDE DIAMETERS OF 6⅝ AND 8⅝ INCHES, IN 20-FOOT LENGTHS. BASED ON TESTS BY PROF. STEWART, 1902–4.

Note that individual experiments are represented by crosses (+), calculated values by circles (○), and group averages by combined circles and crosses (-ọ-).

The scattering of individual results as compared with the general average appears, from these charts, to be restricted to comparatively small bounds when it is considered that we are dealing here with a product that varies noticeably in a number of the characteristics that go to make up its strength. Since these charts represent the results of tests on the common run of commercial lap-welded Bessemer-steel tubes, taken at random from the stock, it is surprising that the scattering of individual results is not greater than that shown.

Apparent Fiber Stress on Wall of Tube at Instant of Collapse.

Fig. 51 shows the apparent compressive stress, in pounds per square inch, at the instant of collapse, on the walls of the tubes constituting Series Two. This chart is constructed to a horizontal scale representing thickness of wall divided by outside diameter of tube and a vertical scale representing apparent fiber stress in pounds per square inch.

The crosses (+) represent the apparent fiber stress of the group averages of Figs. 43-46, the attached figures indicating the outside diameter of tube, while the curve represents the formulæ E and F, plotted to the same scales. These formulæ, which were deduced to represent the most probable values of the apparent fiber stress in the walls of the tubes constituting Series Two, at instant of collapse, are as follows:

For values of $\frac{t}{d}$ less than 0.023:

$$S = 500\,\frac{d}{t}\left(1 - \sqrt{1 - 1{,}600\,\frac{t^2}{d^2}}\right) \quad . \quad . \quad . \quad . \quad \text{(E)}$$

And for values of $\frac{t}{d}$ greater than 0.023:

$$S = 43{,}335 - 693\,\frac{d}{t} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(F)}$$

Where S = apparent fiber stress in lbs. per sq. inch,
d = outside diameter of tube in inches,
t = thickness of wall in inches.

An inspection of this chart will show that the apparent fiber stress on the wall of the tube at instant of collapse varied all the way from about 7,000 pounds per square inch for the relatively

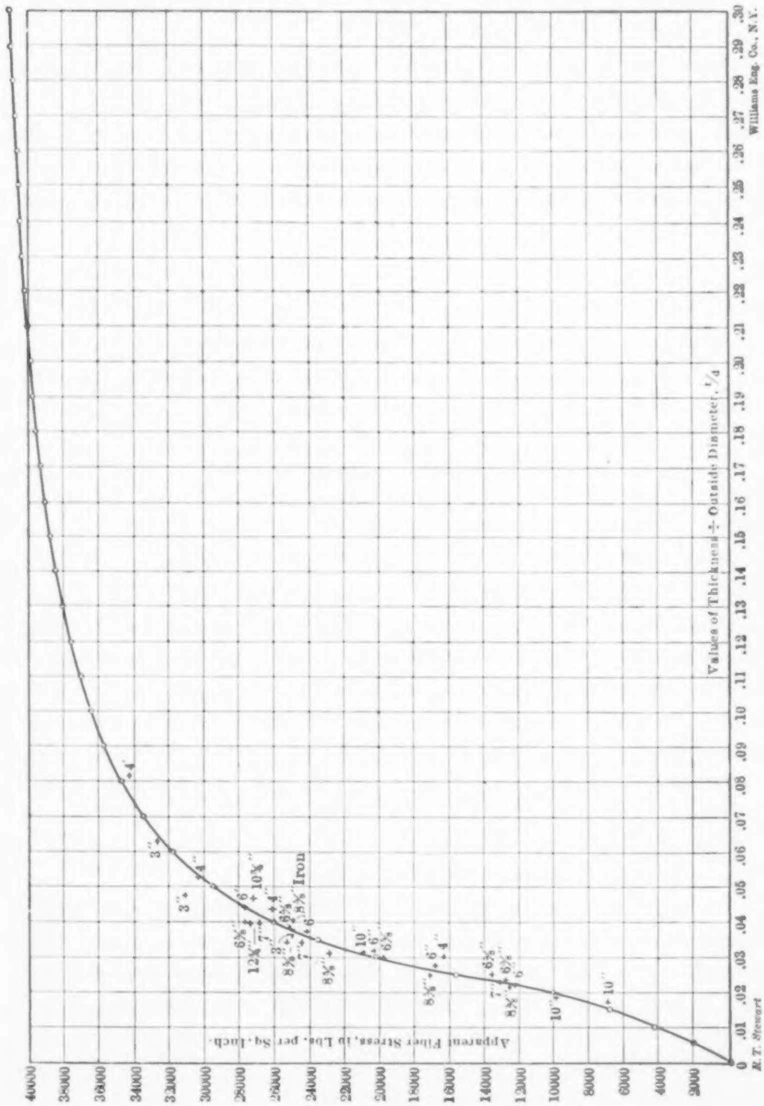


FIG. 51.—CHART SHOWING ACTUAL AND CALCULATED APPARENT FIBER STRESS ON WALL OF TUBE AT INSTANT OF COLLAPSE, PLOTTED TO THICKNESS ÷ DIAMETER, $^t/_d$, FOR NATIONAL TUBE CO.'S BESSEMER-STEEL LAP-WELDED TUBES. BASED ON TESTS ON 20-FOOT LENGTHS BY PROF. STEWART, 1902–4.

Note that crosses (+) represent group averages of tests, the attached figures indicating outside diameters, while circles (○) represent calculated values.

thinnest to 35,000 pounds per square inch for the relatively thickest walls.

This chart shows conclusively that the ability of a commercial wrought tube to withstand a fluid-collapsing pressure is not dependent alone upon either the ultimate strength or elastic limit of the material constituting it. A study of this chart has led to some very interesting deductions which will be dealt with in a separate paper.

Relation of Point of Collapse to Length of Tube.

Theoretically a tube should begin to callapse at the middle of its length, that is, at a point half way between transverse joints, or end connections, tending to hold it to a circular form. This statement is, of course, based upon the assumption that the material of the tube is perfectly homogeneous in its physical properties and that the diameter and thickness of wall are strictly constant throughout its entire length.

The truth of the above statement becomes apparent when we consider that the strength of a tube to resist collapsing pressure depends upon, first, the transverse rigidity of its wall and, second, the tendency of the end connections to hold the tube to a circular form. Since the former, for the assumptions made, would be constant from end to end of the tube and since the latter tendency would become less as the distance from an end connection increases, it is evident that a theoretically perfect tube subjected to a fluid-collapsing pressure would be weakest at a point that is at the greatest possible distance from both of its ends, which point is, of course, located at the middle of its length.

In commercial tubes, however, the material is not strictly homogeneous in its physical properties and there is also a slight variation in out-of-roundness of the different cross-sections, from end to end, as well as a perceptible variation in thickness of wall. Because of these a commercial tube is not necessarily weakest against collapsing pressure at the middle point of its length, as is the case for the theoretically perfect tube.

The actual relation of the point of collapse to the length of tube, for the several hundred commercial tubes tested, is shown in Fig. 52. This chart represents a 20-foot tube divided into foot lengths and numbered consecutively, beginning at the left-hand end. Over each division is placed the Log number of the experimental tubes that collapsed at points nearest to that division.

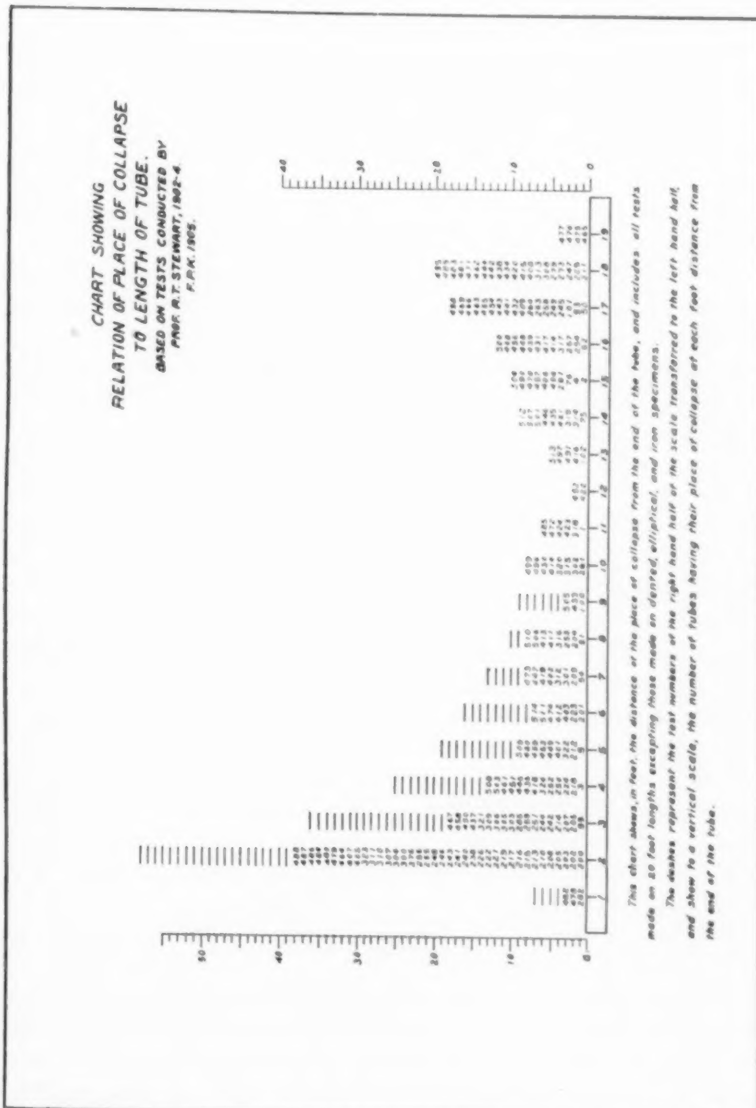


CHART SHOWING
RELATION OF PLACE OF COLLAPSE
TO LENGTH OF TUBE.
BASED ON TESTS CONDUCTED BY
PROF. R.T. STEWART, 1902-4.
F.P.K. 1905.
This chart shows, in feet, the distance of the place of collapse from the end of the tube, and includes all tests made on 20 foot lengths excepting those made on dented, elliptical, and iron specimens.
The dashes represent the test numbers of the right hand half of the scale transferred to the left hand half, and show to a vertical scale, the number of tubes having their place of collapse at each foot distance from the end of the tube.

FIG. 52.

Thus, experimental tubes Nos. 100, 433 and 505 collapsed nearest the 9-foot division from the left-hand end of tube, while Nos. 422 and 452 collapsed nearest the 12-foot division.

It will be observed that the greater number of the tubes collapsed at points that are at distances of 2 feet and 18 feet from the left-hand end, that is, at a distance of 2 feet from either end, while comparatively few collapsed at or near the middle of their lengths. In fact, this chart shows that more than seven times as many of the experimental tubes collapsed at two feet from either end than at a point midway between the ends.

In order to have this chart show the relation of the point of collapse to the nearest end of the tube, it is obvious that we should transfer the test numbers of the right-hand half to the corresponding columns of the left-hand half; for example, we should transfer the test numbers over division 18, which is two feet from the right-hand end, to the column over division 2.

This has been done for all the columns of the right-hand half of the chart, the dashes shown being made to represent the test numbers of the right-hand half of the scale transferred to the corresponding columns of the left-hand half.

Since these experimental tubes were obtained by sending in orders in the usual commercial way, presumably they were taken at random from the company's stock, and, having been handled several times before being placed in the test cylinder, it is obvious that, since it is not known in which direction any of the tubes were passed through the mill while being manufactured, no significance can be attached to the fact that a greater number of the tubes failed nearer the left than the right-hand end. This chart, however, shows very clearly that the bulk of tubes placed under test were least capable of resisting fluid-collapsing pressure at a point near one end.

The reason why the bulk of these tubes collapsed near one end is evidently due chiefly to the following two facts, namely, (1) that a tube subjected to collapsing pressure is weakest at the point where the departure from roundness is greatest, even when this is small, see Fig. 54, and (2) that the greatest departure from roundness for the bulk of these tubes was near one end, see Fig. 53.

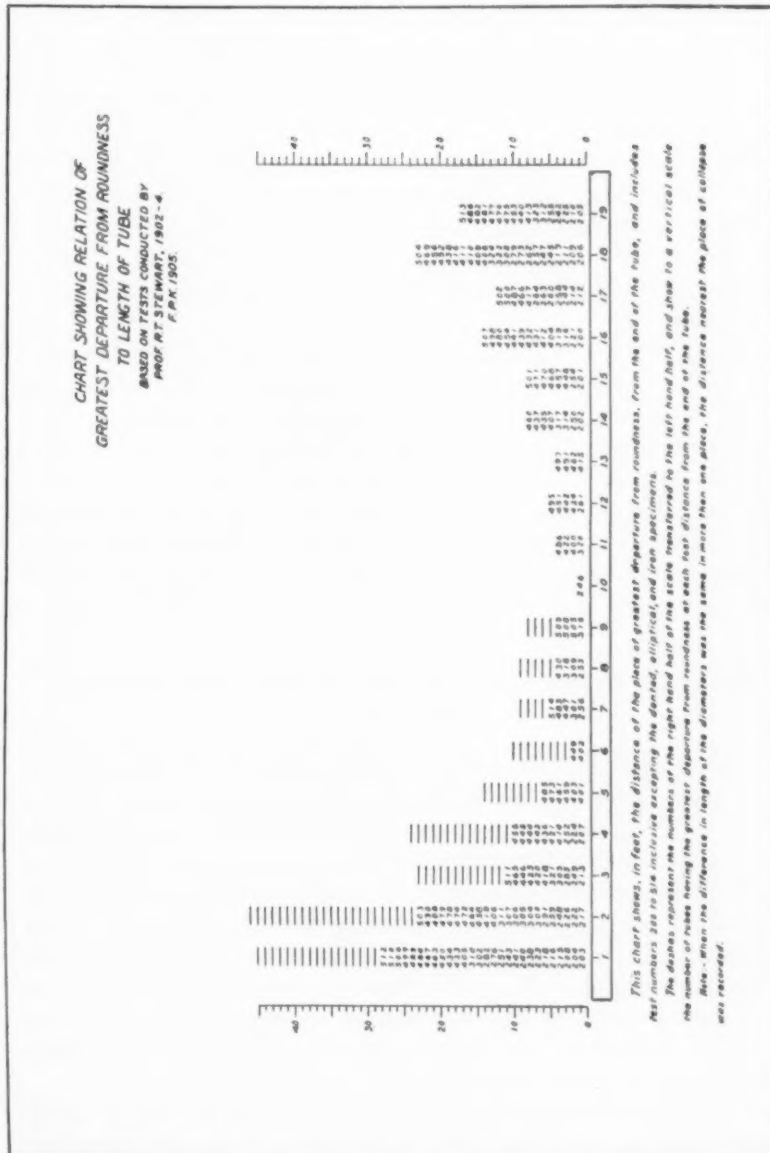


CHART SHOWING RELATION OF
GREATEST DEPARTURE FROM ROUNDNESS
TO LENGTH OF TUBE
BASED ON TESTS CONDUCTED BY
PROF. R.T. STEWART, 1902-4.
F.P.K. 1905.

FIG. 53.

Chart Showing Relation of Greatest Departure from Roundness to Length of Tube.

Fig. 53 shows at a glance how the place of greatest departure from roundness is related to length of tube. For an explanation of the manner of construction see the description of Fig. 52, the two having been constructed according to the same general plan, the only difference being that Fig. 52 shows the location along the length of the tube of the point of collapse, while Fig. 53 shows similarly the location of the point of greatest departure from roundness.

These two charts, taken in connection with Fig. 54, show that the element of greatest weakness in a commercial lap-welded tube is its departure from roundness, even when this departure from roundness is comparatively small, as was the case with the tubes tested. Comparing these three charts with Fig. 56, it will be seen that the thinnest portion of wall, while in itself an element of weakness, is wholly subordinate to out-of-roundness in its influence upon the collapsing strength of commercial lap-welded tubes.

Relation of Axis of Collapse to Smallest Diameter of Tube.

The autographic calipering diagrams taken from the tubes before being placed in the hydraulic test apparatus show, as was to be expected, that none of the tubes tested were exactly round. This departure from roundness, while measurable by the refined methods used for its determination, was, nevertheless, small, varying all the way from zero to as much as possibly 2 per cent. of the diameter. It is apparent that, for homogeneous material and uniform thickness of wall, a tube whose cross-section is not circular will start to yield in the direction of its smallest diameter, and the axis of collapse will be coincident with the original smallest diameter at the place of collapse.

That the slight out-of-roundness of the tubes tested was the chief factor in determining the place of collapse is quite apparent from an inspection of Fig. 54.

This chart shows, for each tube whose test number appears, to the nearest 5 degrees, the angular distance from the axis of col-

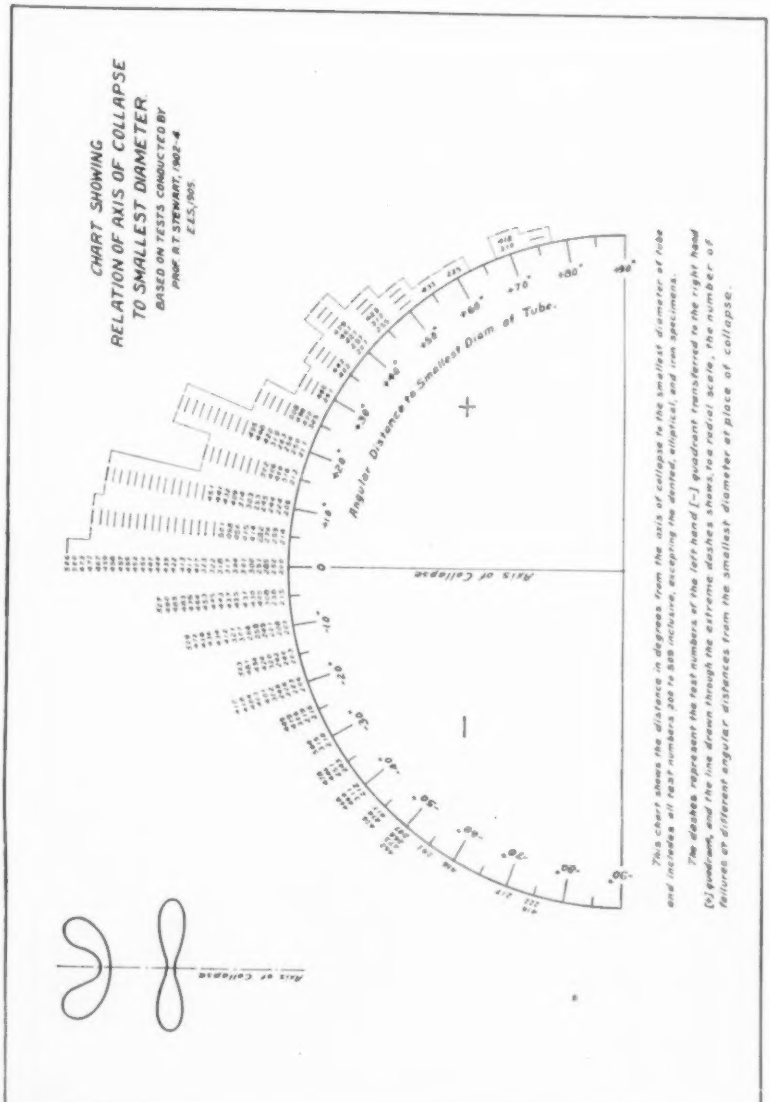


CHART SHOWING
RELATION OF AXIS OF COLLAPSE
TO SMALLEST DIAMETER.
BASED ON TESTS CONDUCTED BY
PROF. R.T. STEWART, 1902-4.
E.L.S. 1905.
Angular Distance to Smallest Diam. of Tube.
Axis of Collapse
Axis of Collapse
This chart shows the distance in degrees from the axis of collapse to the smallest diameter of tube and includes all test numbers 200 to 509 inclusive, excepting the dented, elliptical, and iron specimens.
The dashes represent the test numbers of the left hand [−] quadrant transferred to the right hand [+] quadrant, and the line drawn through the extreme dashes shows, to a radial scale, the number of failures at different angular distances from the smallest diameter at place of collapse.

FIG. 54.

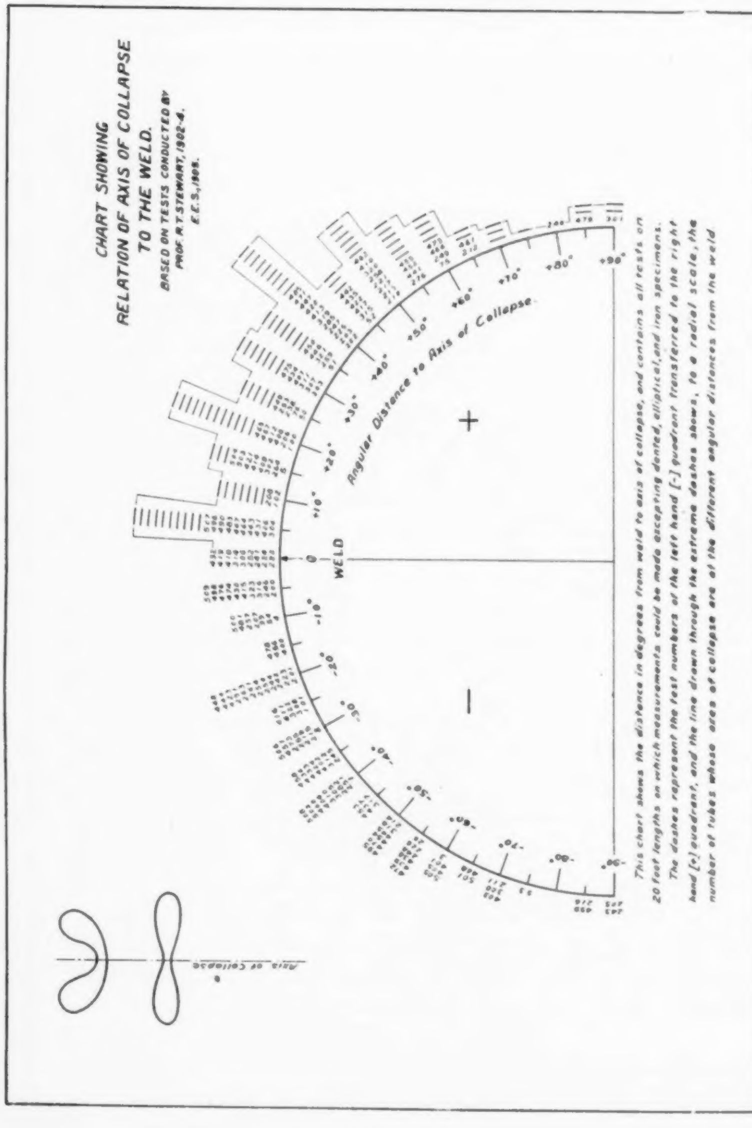


CHART SHOWING
RELATION OF AXIS OF COLLAPSE
TO THE WELD.
BASED ON TESTS CONDUCTED BY
PROF. R.T. STEWART, 1902-4.
E.E.S., 1905.
Angular Distance to Axis of Collapse
WELD
+
−
Axis of Collapse
This chart shows the distance in degrees from weld to axis of collapse, and contains all tests on
20 foot lengths on which measurements could be made excepting dented, elliptical, and iron specimens.
The dashes represent the test numbers of the left hand [-] quadrant transferred to the right
hand [+] quadrant, and the line drawn through the extreme dashes shows, to a radial scale, the
number of tubes whose axes of collapse are at the different angular distances from the weld

Fig. 55.

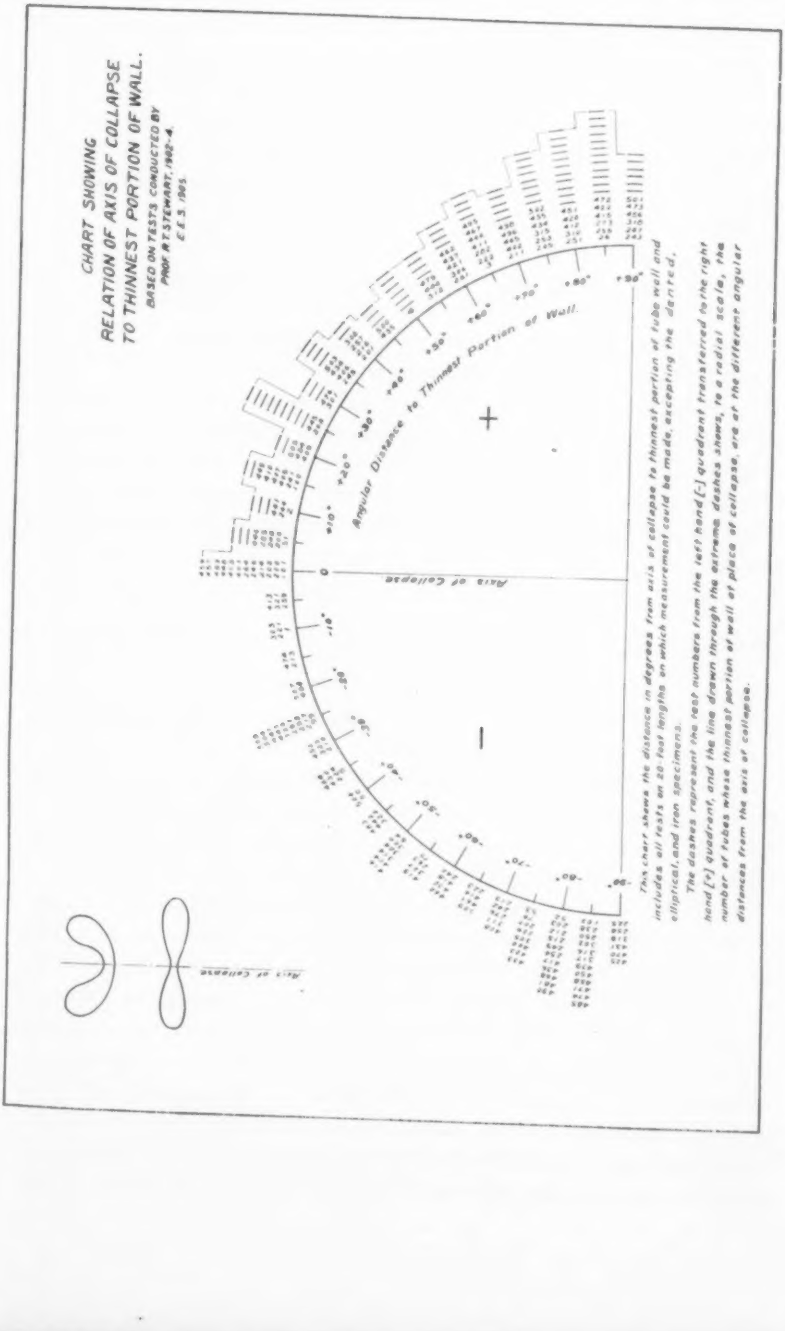


CHART SHOWING
RELATION OF AXIS OF COLLAPSE
TO THINNEST PORTION OF WALL.
BASED ON TESTS CONDUCTED BY
PROF. R.T. STEWART, 1902-4.
C.E.S. 1905.
Angular Distance to Thinnest Portion of Wall.
Axis of Collapse
+
−
0
+10°
+20°
+30°
+40°
+50°
+60°
+70°
+80°
+90°
-10°
-20°
-30°
-40°
-50°
-60°
-70°
-80°
-90°
Axis of Collapse
This chart shows the distance in degrees from axis of collapse to thinnest portion of tube wall and includes all tests on 20-foot lengths on which measurement could be made, excepting the dented, elliptical, and iron specimens.
The dashes represent the test numbers from the left hand [-] quadrant transferred to the right hand [+] quadrant, and the line drawn through the extreme dashes shows, to a radial scale, the number of tubes whose thinnest portion of wall at place of collapse, are at the different angular distances from the axis of collapse.

FIG. 56.

lapse to the nearest end of the orginal smallest diameter of the cross-section through the place of collapse. Since no significance need be attached to the plus and minus signs on this chart, seeing that had any tube been placed in a reversed position in the test apparatus it would have also had the sign of its angular distance from the axis of collapse reversed, the test numbers having negative angles have been transferred to the corresponding columns containing those having positive angles. In order to avoid confusion the places of the test numbers thus transferred are indicated by dashes.

Relation of Axis of Collapse to the Weld.

Fig. 55 is constructed on the same general plan as Fig. 54, for explanation of which see above.

This Chart shows that the angular distance from the weld to the axis of collapse, for the different test numbers, is quite uniformly distributed over about two-thirds of the possible distribution and shows conclusively that the weld, in itself, is not an element of weakness for tubes that are subjected to external fluid pressure.

Relation of Axis of Collapse to the Thinnest Portion of Wall.

Fig. 56 is constructed on the same general plan as Fig. 54, for explanation of which see above.

This chart shows a fairly uniform distribution of the test numbers over about three-fourths of the possible distribution on either side of the axis of collapse, with a somewhat prominent increase over the remaining fourth.

A study of this chart in connection with Fig. 54 will lead to the conclusion that the tendency of commercial tubes is to fail so as to have the axis of collapse at right angles to the diameter through the thinnest portion of the tube. It should be observed in this connection that the bending action on the wall of a tube while being collapsed is most pronounced at this same point, that is to say, at 90 degrees from the axis of collapse. It will also appear, from these same charts, that for commercial lap-welded tubes the usual departure from roundness has a more pronounced effect in determining the manner of collapse. In other words,

when these two influences are related so as to oppose each other the latter almost invariably predominates.

Application to Practice of Stewart's Formulae A and B for the Collapsing Pressures of Lap-welded Steel Tubes.

Table of Collapsing Pressures and Weights.—The probable collapsing pressures contained in the table, Figs. 57 and 58, were calculated by means of formulæ A and B, see page 793.

These formulæ were derived from results of tests on 20-foot lengths of Bessemer-steel lap-welded tubes. They are, however, substantially correct for any length greater than about six diameters of tube between transverse joints or end connections tending to hold the tube to a circular form.

In the columns headed "C.P." are entered the probable collapsing-fluid pressures in pounds per square inch, as calctilated by formulæ A and B; while in columns headed "Wt." are entered the corresponding plain-end weights in pounds per foot length. These weights were calculated on the basis of one cubic inch steel weighing 0.2833 pound. It will be noted that each weight column and the corresponding collapsing-pressure column taken together constitute a double column that is headed by the outside diameter of the tube to which this double column corresponds.

Example 1. Find the plain-end weight and the probable collapsing pressure of a lap-welded Bessemer-steel tube whose outside diameter is 6 inches and thickness of wall 0.180 inch.

In double column headed "6 O.D.," Fig. 58, and opposite 0.18 in the extreme left-hand column read 11.19 and 1214, the first being the required plain-end weight in pounds per foot length and the second the probable collapsing fluid pressure in pounds per square inch. This collapsing pressure is for a 20-foot length between transverse joints or other supports tending to hold the tube to a circular form, but is also substantially correct for any length greater than about 6 diameters or, in this case, 3 feet.

Example 2. Find the collapsing pressure of a tube 7 inches outside diameter having a plain-end weight of 17 pounds per foot.

From the double column head "7 O.D.," Fig. 58, we find that a plain-end weight of 17.33 pounds per foot corresponds to a probable collapsing pressure of 1,586 pounds per square inch, and also that a weight of 16.63 pounds corresponds to a collapsing

TH'K'S WALL, INCHES	2 IN. O.D.		2¼ IN. O.D.		2½ IN. O.D.		2¾ IN. O.D.	
	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.
0.10	2.03	2,947	2.30	2,466	2.56	2,081	2.83	1,766
0.11	2.22	3,381	2.51	2,851	2.81	2,427	3.10	2,081
0.12	2.41	3,814	2.73	3,236	3.05	2,774	3.37	2,396
0.13	2.60	4,248	2.94	3,622	3.29	3,121	3.64	2,711
0.14	2.78	4,681	3.15	4,007	3.53	3,468	3.90	3,026
0.15	2.96	5,114	3.36	4,392	3.76	3,814	4.17	3,341
0.16	3.14	5,548	3.57	4,777	4.00	4,161	4.43	3,657
0.17	3.32	5,981	3.78	5,162	4.23	4,508	4.68	3,972
0.18	3.50	6,414	3.98	5,548	4.46	4,854	4.94	4,287
0.19	3.67	6,848	4.18	5,933	4.69	5,201	5.19	4,602
0.20	3.84	7,281	4.38	6,318	4.91	5,548	5.45	4,917
0.21	4.01	7,714	4.58	6,703	5.14	5,894	5.70	5,232
0.22	4.18	8,148	4.77	7,088	5.36	6,241	5.94	5,548
0.23	4.35	8,581	4.96	7,474	5.58	6,588	6.19	5,863
0.24	4.51	9,014	5.15	7,859	5.79	6,934	6.43	6,178
0.25	4.67	9,448	5.34	8,244	6.01	7,281	6.68	6,493
0.26	4.83	9.881	5.53	8,629	6.22	7,628	6.91	6,808
0.27	—	—	5.71	9,014	6.43	7,974	7.15	7,123
0.28	—	—	5.89	9,400	6.64	8,321	7.39	7,439
0.29	—	—	6.07	9,785	6.84	8,668	7.62	7,754
0.30	—	—	—	—	7.05	9,014	7.85	8,069
0.31	—	—	—	—	7.25	9,361	8.08	8,384
0.32	—	—	—	—	7.45	9,708	8.30	8,699
0.33	—	—	—	—	—	—	8.53	9,014
0.34	—	—	—	—	—	—	8.75	9.330
0.35	—	—	—	—	—	—	8.97	9,645
0.36	—	—	—	—	—	—	9.19	9,960
0.37	—	—	—	—	—	—	—	—
0.38	—	—	—	—	—	—	—	—
0.39	—	—	—	—	—	—	—	—
0.40	—	—	—	—	—	—	—	—
0.41	—	—	—	—	—	—	—	—
0.42	—	—	—	—	—	—	—	—
0.43	—	—	—	—	—	—	—	—
0.44	—	—	—	—	—	—	—	—
0.45	—	—	—	—	—	—	—	—
0.46	—	—	—	—	—	—	—	—
0.47	—	—	—	—	—	—	—	—
0.48	—	—	—	—	—	—	—	—
0.49	—	—	—	—	—	—	—	—
0.50	—	—	—	—	—	—	—	—

TH'K'S WALL, INCHES	3 IN. O.D.		3¼ IN. O.D.		3½ IN. O.D.		3¾ IN. O.D.	
	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.
0.10	3.10	1,503	3.36	1,281	3.63	1,090	3.90	925
0.11	3.40	1,792	3.69	1,547	3.98	1,338	4.28	1,156
0.12	3.69	2,081	4.01	1,814	4.33	1,586	4.65	1,387
0.13	3.98	2,370	4.33	2,081	4.68	1,833	5.03	1,619
0.14	4.28	2,659	4.65	2,347	5.02	2,081	5.40	1,850
0.15	4.57	2,947	4.97	2,614	5.37	2,328	5.77	2,081
0.16	4.85	3,236	5.28	2,881	5.71	2,576	6.13	2,312
0.17	5.14	3,525	5.59	3,148	6.05	2,824	6.50	2,543
0.18	5.42	3,814	5.90	3,414	6.38	3,071	6.86	2,774
0.19	5.70	4,103	6.21	3,681	6.72	3,319	7.22	3,005
0.20	5.98	4,392	6.51	3,948	7.05	3,567	7.58	3,236
0.21	6.26	4,681	6.82	4,214	7.38	3,814	7.94	3,468
0.22	6.53	4,970	7.12	4,481	7.71	4,062	8.29	3,699
0.23	6.80	5,259	7.42	4,748	8.03	4,309	8.65	3,930
0.24	7.07	5,548	7.72	5,014	8.36	4,557	9.00	4,161
0.25	7.34	5,836	8.01	5,281	8.68	4,805	9.35	4,392
0.26	7.61	6,125	8.30	5,548	9.00	5,052	9.69	4,623
0.27	7.87	6,414	8.59	5,814	9.31	5,300	10.04	4,854
0.28	8.13	6,703	8.88	6,081	9.63	5,548	10.38	5,085
0.29	8.39	6,992	9.17	6,348	9.94	5,795	10.72	5,316
0.30	8.65	7,281	9.45	6,614	10.25	6,043	11.05	5,548
0.31	8.91	7,570	9.73	6,881	10.56	6,290	11.39	5,779
0.32	9.16	7,859	10.01	7,148	10.87	6,538	11.72	6,010
0.33	9.41	8,148	10.29	7,414	11.17	6,786	12.05	6,241
0.34	9.66	8,437	10.57	7,681	11.47	7,033	12.38	6,472
0.35	9.91	8,725	10.84	7,948	11.77	7,281	12.71	6,703
0.36	10.15	9,014	11.11	8,214	12.07	7,529	13.03	6,934
0.37	10.39	9,303	11.38	8,481	12.37	7,776	13.36	7,165
0.38	10.63	9,592	11.65	8,748	12.66	8,024	13.68	7,397
0.39	10.87	9,881	11.91	9,014	12.95	8,272	14.00	7,628
0.40	—	—	12.18	9,281	13.24	8,519	14.31	7,859
0.41	—	—	12.44	9,548	13.53	8,767	14.63	8,090
0.42	—	—	12.69	9,814	13.82	9,014	14.94	8,321
0.43	—	—	—	—	14.10	9,262	15.25	8,552
0.44	—	—	—	—	14.38	9,510	15.55	8,783
0.45	—	—	—	—	14.66	9,757	15.86	9,014
0.46	—	—	—	—	—	—	16.16	9,246
0.47	—	—	—	—	—	—	16.46	9,477
0.48	—	—	—	—	—	—	16.76	9,708
0.49	—	—	—	—	—	—	17.06	9,939
0.50	—	—	—	—	—	—	—	—

TH'K'S WALL, INCHES	4 IN. O.D.		4¼ IN. O.D.		4½ IN. O.D.		4¾ IN. O.D.	
	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.
0.10	4.17	781	4.43	653	4.70	542	4.97	461
0.11	4.57	997	4.86	857	5.16	733	5.45	621
0.12	4.97	1,214	5.29	1,061	5.61	925	5.93	804
0.13	5.37	1,431	5.72	1,265	6.07	1,118	6.41	986
0.14	5.77	1,647	6.15	1,469	6.52	1,310	6.89	1.168
0.15	6.17	1,864	6.57	1,673	6.97	1,503	7.37	1,351
0.16	6.56	2,081	6.99	1,877	7.42	1,696	7.84	1,533
0.17	6.95	2,297	7.41	2,081	7.86	1,888	8.32	1,716
0.18	7.34	2,514	7.82	2,285	8.30	2,081	8.79	1,898
0.19	7.73	2,731	8.24	2,489	8.75	2,273	9.25	2,081
0.20	8.12	2,947	8.65	2,693	9.18	2,466	9.72	2,263
0.21	8.50	3,164	9.06	2,897	9.62	2,659	10.18	2,446
0.22	8.88	3,381	9.47	3,100	10.06	2,851	10.64	2,628
0.23	9.26	3,598	9.87	3,304	10.49	3,044	11.40	2,811
0.24	9.64	3,814	10.28	3,508	10.92	3,236	11.56	2,993
0.25	10.01	4,031	10.68	3,712	11.35	3,429	12.02	3,176
0.26	10.39	4,248	11.08	3,916	11.77	3,622	12.47	3,358
0.27	10.76	4,464	11.48	4,120	12.20	3,814	12.92	3,541
0.28	11.12	4,681	11.87	4,324	12.62	4,007	13.37	3,723
0.29	11.49	4,898	12.27	4,528	13.04	4,199	13.81	3,905
0.30	11.85	5,114	12.66	4,732	13.46	4,392	14.26	4,088
0.31	12.22	5,331	13.04	4,936	13.87	4,585	14.70	4,270
0.32	12.58	5,548	13.43	5,140	14.29	4,777	15.14	4,453
0.33	12.93	5,764	13.82	5,344	14.70	4,970	15.58	4,635
0.34	13.29	5,981	14.20	5,548	15.11	5,162	16.01	4,818
0.35	13.64	6.198	14.58	5,752	15.51	5,355	16.45	5,000
0.36	14.00	6,414	14.96	5,955	15.92	5,548	16.88	5,183
0.37	14.34	6,631	15.33	6,159	16.32	5,740	17.31	5,365
0.38	14.69	6,848	15.71	6.363	16.72	5,933	17.74	5,548
0.39	15.04	7,064	16.08	6,567	17.12	6,125	18.16	5,730
0.40	15.38	7,281	16.45	6,771	17.52	6,318	18.58	5,913
0.41	15.72	7,498	16.81	6,975	17.91	6,511	19.00	6,095
0.42	16.06	7,714	17.18	7,179	18.30	6,703	19.42	6,277
0.43	16.40	7,931	17.54	7,383	18.69	6,896	19.84	6,460
0.44	16.73	8,148	17.90	7,587	19.08	7,088	20.25	6,642
0.45	17.06	8,364	18.26	7,791	19.46	7,281	20.67	6,825
0.46	17.39	8,681	18.62	7,995	19.85	7,474	21.08	7,007
0.47	17.72	8,798	18.97	8,199	20.23	7,666	21.48	7,190
0.48	18.05	9,014	19.33	8,403	20.61	7,859	21.89	7,372
0.49	18.37	9,231	19.68	8,607	20.99	8,051	22.29	7,555
0.50	18.69	9,448	20.03	8,810	21.36	8,244	22.70	7,737

FIG. 57.—COLLAPSING PRESSURES OF LAP-WELDED STEEL TUBES, CALCULATED BY PROF. STEWART'S FORMULÆ A AND B.

THK'S WALL INCHES	5 IN. O.D.		5¼ IN. O.D.		5½ IN. O.D.		6 IN. O.D.		6⅝ IN. O.D.		7 IN. O.D.		7⅝ IN. O.D.		8 IN. O.D.		8⅝ IN. O.D.		9 IN. O.D.		10 IN. O.D.		11 IN. O.D.	
	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.	WT.	C.P.
0.10																								
0.11	5.74	525	6.04	454	6.33	400																		
0.12	6.25	694	6.57	595	6.90	512	7.54	400	8.34	311														
0.13	6.76	867	7.11	760	7.46	663	8.15	501	9.02	380	9.54	331	10.41	269	10.93	240								
0.14	7.27	1,041	7.64	925	8.01	820	8.76	636	9.70	466	10.26	400	11.19	321	11.75	286	12.69	239	13.25	217				
0.15	7.77	1,214	8.17	1,090	8.57	978	9.37	781	10.37	576	10.97	485	11.98	383	12.58	339	13.58	282	14.18	255	15.78	200		
0.16	8.27	1,387	8.70	1,255	9.13	1,135	9.98	925	11.05	707	11.69	595	12.76	456	13.40	400	14.47	330	15.11	297	16.81	232	18.52	187
0.17	8.77	1,561	9.22	1,420	9.68	1,293	10.59	1,070	11.72	838	12.40	719	13.54	548	14.22	473	15.35	385	16.03	345	17.85	267	19.66	214
0.18	9.27	1,734	9.75	1,586	10.23	1,450	11.19	1,214	12.39	969	13.11	843	14.31	660	15.03	564	16.23	449	16.96	400	18.88	306	20.80	244
0.19	9.76	1,907	10.27	1,751	10.78	1,608	11.79	1,359	13.06	1,100	13.82	966	15.09	774	15.85	672	17.12	527	17.88	464	19.91	350	21.94	277
0.20	10.25	2,081	10.79	1,916	11.32	1,766	12.39	1,503	13.72	1,230	14.53	1,090	15.86	887	16.66	781	18.00	624	18.80	542	20.93	400	23.07	314
0.21	10.74	2,254	11.30	2,081	11.86	1,923	12.99	1,647	14.39	1,361	15.23	1,214	16.63	1,001	17.47	889	18.87	724	19.71	636	21.96	457	24.20	354
0.22	11.23	2,427	11.82	2,246	12.41	2,081	13.58	1,792	15.05	1,492	15.93	1,338	17.40	1,115	18.28	997	19.75	825	20.63	733	22.98	525	25.33	400
0.23	11.72	2,601	12.33	2,411	12.95	2,238	14.17	1,936	15.71	1,623	16.63	1,462	18.17	1,228	19.09	1,106	20.62	925	21.54	829	24.00	607	26.46	452
0.24	12.20	2,774	12.84	2,576	13.48	2,396	14.76	2,081	16.37	1,754	17.33	1,586	18.93	1,342	19.89	1,214	21.49	1,026	22.45	925	25.02	694	27.58	512
0.25	12.68	2,947	13.35	2,741	14.02	2,554	15.35	2,225	17.02	1,885	18.02	1,709	19.69	1,456	20.69	1,322	22.36	1,126	23.36	1,022	26.03	781	28.70	584
0.26	13.16	3,121	13.86	2,906	14.55	2,711	15.94	2,370	17.67	2,015	18.72	1,833	20.45	1,569	21.49	1,431	23.23	1,227	24.27	1,118	27.05	867	29.82	663
0.27	13.64	3,294	14.36	3,071	15.08	2,869	16.52	2,514	18.33	2,146	19.41	1,957	21.21	1,683	22.29	1,539	24.09	1,327	25.17	1,214	28.06	954	30.94	741
0.28	14.11	3,468	14.86	3,236	15.61	3,026	17.11	2,659	18.97	2,277	20.10	2,081	21.96	1,797	23.09	1,647	24.96	1,428	26.08	1,310	29.07	1,041	32.06	820
0.29	14.59	3,641	15.36	3,401	16.14	3,184	17.69	2,803	19.62	2,408	20.78	2,205	22.72	1,910	23.88	1,756	25.82	1,528	26.98	1,407	30.07	1,127	33.17	899
0.30	15.06	3,814	15.86	3,567	16.66	3,341	18.26	2,947	20.27	2,539	21.47	2,328	23.47	2,024	24.67	1,864	26.67	1,629	27.88	1,503	31.08	1,214	34.28	978
0.31	15.53	3,988	16.36	3,732	17.18	3,499	18.84	3,092	20.91	2,670	22.15	2,452	24.22	2,138	25.46	1,972	27.53	1,729	28.77	1,599	32.08	1,301	35.39	1,057
0.32	15.99	4,161	16.85	3,897	17.70	3,657	19.41	3,236	21.55	2,800	22.83	2,576	24.97	2,251	26.25	2,081	28.38	1,830	29.67	1,696	33.08	1,387	36.50	1,135
0.33	16.46	4,334	17.34	4,062	18.22	3,814	19.98	3,381	22.19	2,931	23.51	2,700	25.71	2,365	27.03	2,189	29.24	1,930	30.56	1,792	34.08	1,474	37.61	1,214
0.34	16.92	4,508	17.83	4,227	18.74	3,972	20.55	3,525	22.82	3,062	24.18	2,824	26.45	2,479	27.82	2,297	30.08	2,031	31.45	1,888	35.08	1,561	38.71	1,293
0.35	17.38	4,681	18.32	4,392	19.25	4,129	21.12	3,670	23.46	3,193	24.86	2,947	27.19	2,592	28.60	2,406	30.93	2,131	32.33	1,985	36.07	1,647	39.81	1,372
0.36	17.84	4,854	18.80	4,557	19.76	4,287	21.68	3,814	24.09	3,324	25.53	3,071	27.93	2,706	29.37	2,514	31.78	2,232	33.22	2,081	37.06	1,734	40.91	1,450
0.37	18.30	5,028	19.28	4,722	20.27	4,445	22.25	3,959	24.72	3,454	26.20	3,195	28.67	2,820	30.15	2,622	32.62	2,332	34.10	2,177	38.05	1,821	42.01	1,529
0.38	18.75	5,201	19.76	4,887	20.78	4,602	22.81	4,103	25.35	3,585	26.87	3,319	29.40	2,933	30.93	2,731	33.46	2,433	34.98	2,273	39.04	1,907	43.10	1,608
0.39	19.20	5,374	20.24	5,052	21.28	4,760	23.37	4,248	25.97	3,716	27.53	3,443	30.14	3,047	31.70	2,839	34.30	2,533	35.86	2,370	40.03	1,994	44.19	1,687
0.40	19.65	5,548	20.72	5,217	21.79	4,917	23.92	4,392	26.59	3,847	28.20	3,567	30.87	3,161	32.47	2,947	35.14	2,633	36.74	2,466	41.01	2,081	45.28	1,766
0.41	20.10	5,721	21.19	5,383	22.29	5,075	24.48	4,536	27.21	3,978	28.86	3,690	31.59	3,274	33.24	3,056	35.97	2,734	37.61	2,562	41.99	2,167	46.37	1,844
0.42	20.54	5,894	21.67	5,548	22.79	5,232	25.03	4,681	27.83	4,109	29.52	3,814	32.32	3,388	34.00	3,164	36.80	2,834	38.49	2,659	42.97	2,254	47.46	1,923
0.43	20.99	6,068	22.14	5,713	23.28	5,390	25.58	4,825	28.45	4,239	30.17	3,938	33.04	3,502	34.76	3,273	37.64	2,935	39.36	2,755	43.95	2,341	48.54	2,002
0.44	21.43	6,241	22.60	5,878	23.78	5,548	26.13	4,970	29.06	4,370	30.83	4,062	33.76	3,615	35.53	3,381	38.46	3,035	40.23	2,851	44.93	2,427	49.62	2,081
0.45	21.87	6,414	23.07	6,043	24.27	5,705	26.67	5,114	29.68	4,501	31.48	4,186	34.48	3,729	36.29	3,489	39.29	3,136	41.09	2,947	45.90	2,514	50.70	2,160
0.46	22.30	6,588	23.53	6,208	24.76	5,863	27.22	5,259	30.29	4,632	32.13	4,309	35.20	3,843	37.04	3,598	40.11	3,236	41.96	3,044	46.87	2,601	51.78	2,238
0.47	22.74	6,761	23.99	6,373	25.25	6,020	27.76	5,403	30.90	4,763	32.78	4,433	35.92	3,956	37.80	3,706	40.94	3,337	42.82	3,140	47.84	2,687	52.86	2,317
0.48	23.17	6,934	24.45	6,538	25.73	6,178	28.30	5,548	31.50	4,893	33.42	4,557	36.63	4,070	38.55	3,814	41.76	3,437	43.68	3,236	48.80	2,774	53.93	2,396
0.49	23.60	7,108	24.91	6,703	26.22	6,336	28.84	5,692	32.11	5,024	34.07	4,681	37.34	4,184	39.30	3,923	42.57	3,538	44.54	3,333	49.77	2,861	55.00	2,475
0.50	24.03	7,281	25.37	6,868	26.70	6,493	29.37	5,837	32.71	5,165	34.71	4,805	38.05	4,297	40.05	4,031	43.39	3,638	45.39	3,429	50.73	2,947	56.07	2,554

FIG. 58.—COLLAPSING PRESSURES OF STEEL TUBES.—*Continued.*

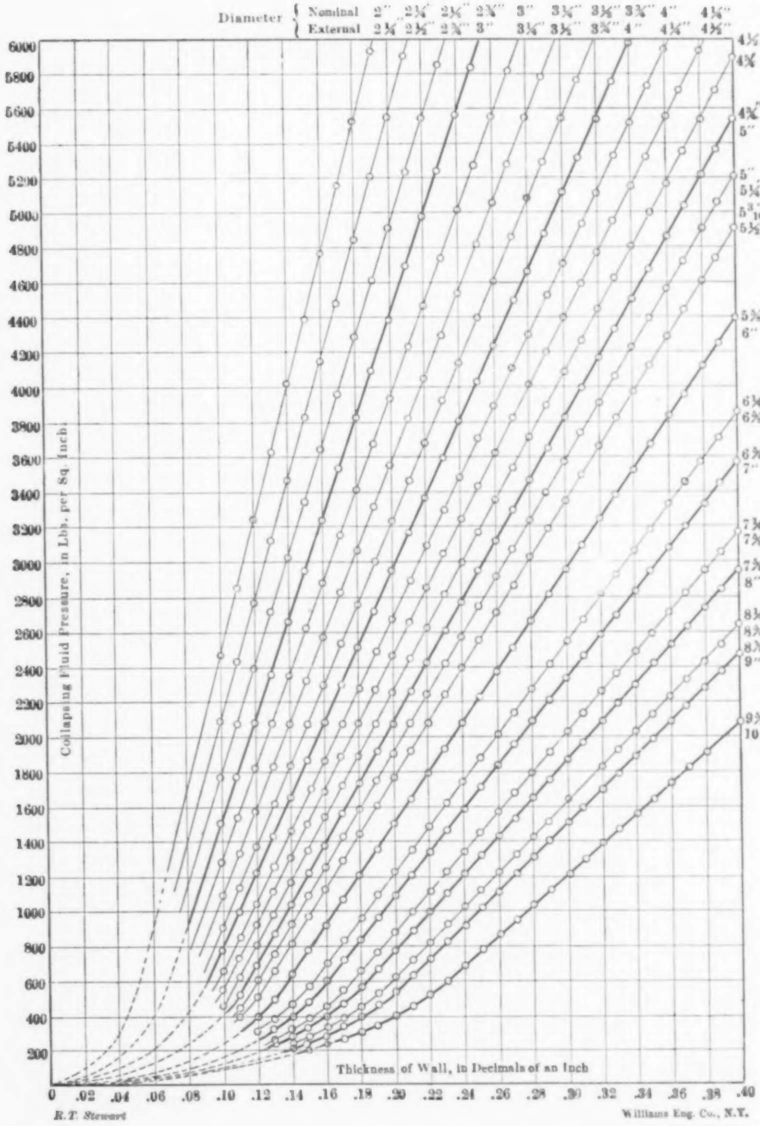


FIG. 59.—CHART SHOWING THE VALUES OF THE TABLE OF COLLAPSING PRESSURES OF LAP-WELDED STEEL TUBES, FIGS. 57 AND 58, CONSTRUCTED TO A VERTICAL SCALE OF COLLAPSING PRESSURES AND A HORIZONTAL SCALE OF THICKNESS OF WALL.

pressure of 1,462 pounds. Now, by the usual method for interpolating it will be found that for a plain-end weight of 17 pounds per foot the corresponding collapsing pressure will be 1,527 pounds per square inch.

It is believed, however, that the tabular values in this table are sufficiently numerous to render it unnecessary to make any interpolations whatever while applying it to practice.

Factors of Safety.—It must be remembered that these tabular values represent the probable collapsing pressures as based upon the tests. This being the case, any individual tube is as likely to fail above as below this most probable pressure. The relation of the collapsing pressure of each individual tube to the most probable, as tabulated, is clearly shown in Figs. 48-50, where the curves represent the tabular values, crosses the collapsing pressures of individual tubes, and combined crosses and circles the adjusted group averages. Expressed in per cent., this variation of each individual collapsing pressure from the tabular is shown in column 9 of Figs. 43-46. This table shows that not one of the several hundred tubes tested failed at a pressure lower than 42 per cent. of the probable collapsing pressure, while ½ of one per cent. of the number of tubes failed at 37 per cent. and 2 per cent. at 25 per cent. of that pressure. In other words, with an actual factor of safety of 1.75, as based upon this table, Figs. 57 and 58, not one of the tubes tested would have failed.

From an inspection of the charts and table above referred to it would appear that:

1. For the most favorable practical conditions, namely, when the tube is subjected only to stress due to fluid pressure and only the most trivial loss could result from its failure, a factor of safety of three would appear sufficient.

2. When only a moderate amount of loss could result from failure use a factor of four.

3. When considerable damage to property and loss of life might result from a failure of the tube, then use a factor of safety of at least six.

4. When the conditions of service are such as to cause the tube to become less capable of resisting collapsing pressure, such as the thinning of wall due to corrosion, the weakening of the material due to over-heating, the creating of internal stress in the wall of the tube due to unequal heating, vibration, etc., the above factors

of safety should be increased in proportion to the severity of these actions.

Example 3. By means of the table, Fig. 57, find what thickness of wall a 4-inch boiler tube should have in order to withstand a working pressure of 200 pounds per square inch, with a factor of safety of eight.

In this case the probable collapsing pressure should equal the working pressure multiplied by the factor of safety, or 1,600 pounds. Now, looking in double column headed "4 O.D.," we find the nearest tabular collapsing pressure to be 1,647 pounds. This corresponds to a thickness of 0.14 inch or No. 9 B.W.G., as read opposite in the extreme left-hand column.

Example 4. Find the plain-end weight per foot of a $6\frac{1}{4}$-inch casing to withstand a maximum difference between external and internal fluid pressures corresponding to a water head of 800 feet, on the basis of a factor of safety of four.

A table of hydrostatic pressures will show that this head of 800 feet will create a fluid pressure of 347 pounds per square inch, tending to collapse the tube at its lower end. Multiplying this by the factor of safety we get 1,388 pounds per square inch as the probable collapsing pressure. Now, looking in double column headed "$6\frac{5}{8}$," which is the outside diameter of a nominal $6\frac{1}{4}$ casing, we find the nearest tabular collapsing pressure to be 1,361 pounds, which corresponds to a plain-end weight of 14.39 pounds and a thickness of wall of 0.21 inch.

Chart Showing Relation of Collapsing Pressure to $\frac{t}{d}$.—Fig. 60 resulted from plotting equations A and B (see pp. 793-795) to a vertical scale of probable collapsing pressures and a horizontal scale representing the thickness of the tube divided by its outside diameter, or the $\frac{t}{d}$ contained in these formulæ.

By plotting in this manner, a single line may be made to represent the collapsing pressures of a great variety of tubes, irrespective of their individual diameters or thicknesses of wall.

It will be noticed that this is the same curve as that shown in Fig. 47, the difference being that it is drawn to larger and more conveniently read scales.

In order to condense the size of this chart the curve is broken into the two parts XX and YY. By this means the area of the chart has been reduced to about one-fourth of that which would

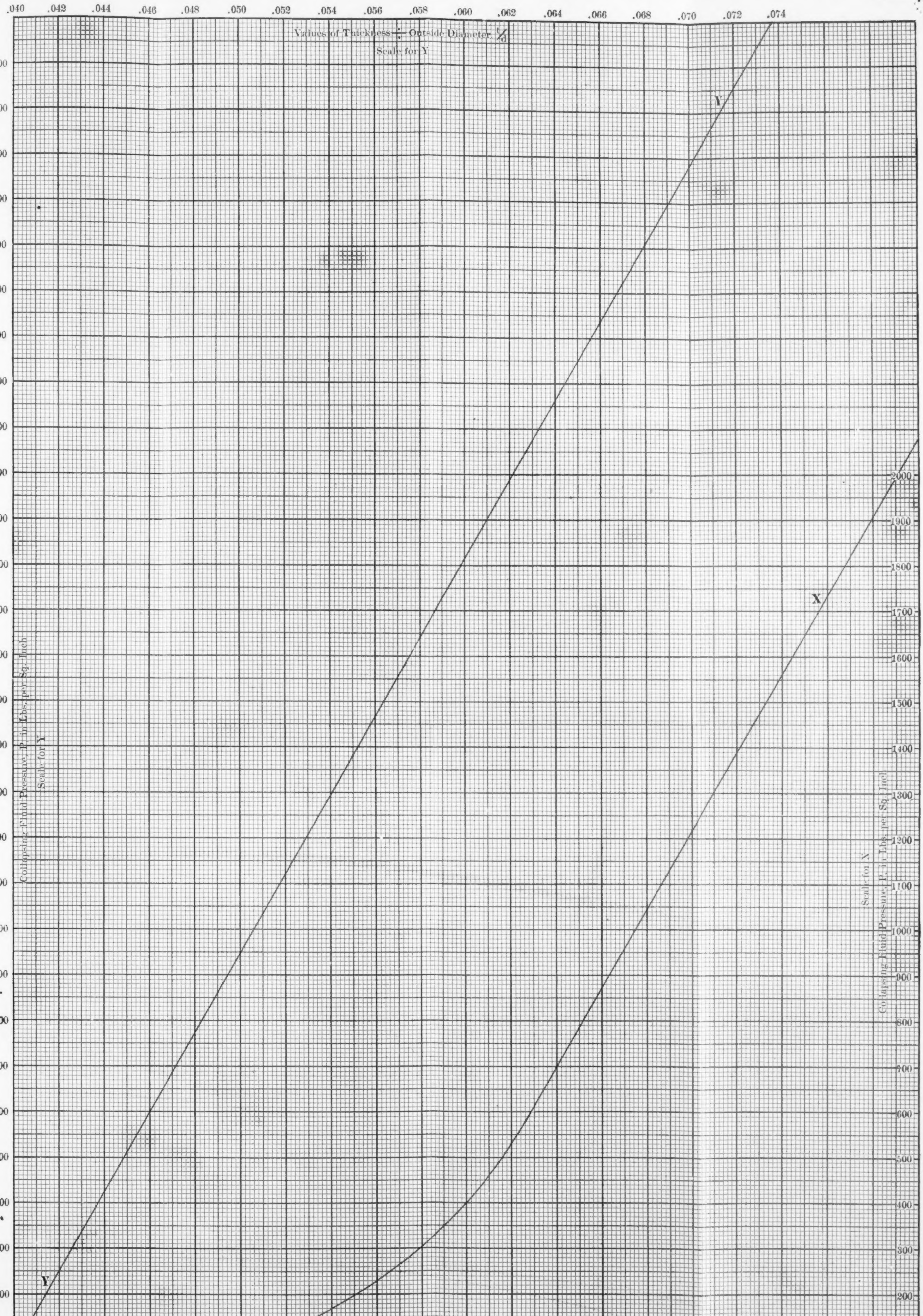


FIG. 60.—CHART FOR OBTAINING COLLAPSING PRESSURES OF LAP-WELDED STEEL TUBES, WHEN THICKNESS AND OUTSIDE DIAMETER ARE GIVEN. BASED ON PROF. STEWART'S FORMULÆ A AND B.
FOR EXPLANATION OF USE, SEE PAGE 816.

otherwise have been required to construct the chart to the scales shown. It will be observed that YY is the upper portion of XX transferred to the left and then dropped down, the break in the curve corresponding to a collapsing pressure of 2,080 pounds and a thickness divided by diameter of 0.040. It will also be observed that the scales for the portion XX are at the lower and right-hand margins, while those for the portion YY are at the upper and left-hand margins.

The smallest divisions on the vertical scale represent 10 pounds collapsing pressure, while those on the horizontal scale represent 0.0002 thickness divided by outside diameter. When reading to the nearest smallest division on these scales the error will not exceed five pounds for probable collapsing pressure, nor 0.0001 for thickness divided by outside diameter.

This, then, is a universal chart showing the relation of the probable collapsing pressure of a tube to the thickness of wall divided by outside diameter. It represents the adjusted values of the group averages of all the 20-foot lengths of the Bessemer steel lap-welded tubes tested, omitting the three that proved to be defective, and may therefore be used with entire confidence within the range of these experiments; that is, for Bessemer steel lap-welded tubes from 2 to 12 inches outside diameter, and for all commercial thickness of wall in lengths greater than about six diameters of tube between joints or end connections tending to hold them to a circular form.

Example 5. Find by means of Fig. 60 the probable collapsing pressure of a tube having an external diameter equal to 6 inches, and a thickness of wall equal to 0.203 inch.

Dividing the thickness of wall by the outside diameter we get $\frac{t}{d}$ equal 0.0338. Since this value is less than 0.04 we look for it on the scale at the lower margin of the chart. Having found it on this scale, look along the vertical line through it until the line XX is reached; then look along the nearest horizontal line toward the right and read from the scale of probable collapsing pressures 1,540 pounds per square inch. This is the probable collapsing pressure for a length of 20 feet, but is also substantially correct for any length greater than about six diameters, or 3 feet for a 6-inch tube, between transverse joints tending to hold the tube to a circular form.

Linear Units for d and t.—It should be noted that both the out-

side diameter, d, and the thickness of wall, t, must be expressed in the same linear unit of measure, as, for example, in inches, centimeters, millimeters, etc. The name of the linear unit is immaterial, the chart being just as applicable to obtaining probable collapsing pressures in pounds per square inch when the diameter and thickness are expressed in metric as when in English units.

Collapsing Pressures in Metric Measure.—First divide the thickness of wall, t, by the outside diameter, d, both being expressed in either inches or millimeters. Second, obtain from Fig. 60, as in example 5, the probable collapsing pressure in pounds per square inch. Third, reduce the resulting collapsing pressure in pounds per square inch to that expressed in kilograms per square centimeter by multiplying by the conversion factor 0.0703.

Example 6. Find the probable collapsing pressure of a tube whose outside diameter and thickness of wall are respectively 15 centimeters and 4 millimeters.

Fifteen centimeters being equal to 150 millimeters, $\frac{t}{d}$, or thickness divided by outside diameter, equals 0.0266. Proceeding as in example 5, we find the probable collapsing pressure to be 920 pounds per square inch. Multiplying this by the conversion factor for reducing English to metric units, given above, we get 920 multiplied by 0.0703, or 64.7 kilograms per square centimeter.

Example 7. With a factor of safety of eight find what thickness a 3-inch boiler tube should have in order to resist a working external fluid pressure of 220 pounds per square inch.

In accordance with these assumptions the probable collapsing pressure of the tube should equal the working pressure multiplied by the factor of safety, or 1,760 pounds per square inch. From Fig. 60 find 1,760 on the scale of probable collapsing pressures at the right-hand margin, and look along the horizontal line through this point until line XX is reached; then look down the nearest vertical line and read 0.0363 as the value of $\frac{t}{d}$ or thickness divided by outside diameter. We can now get the required thickness of wall by multiplying the value of $\frac{t}{d}$ by d, which gives us t equal 0.0363×3, or 0.109 inch, or No. 12 B.W.G.

For the same conditions of pressure, a tube 8 centimeters, or 80 millimeters, diameter should have a thickness of wall equal 0.0363×80, or 2.9 millimeters.

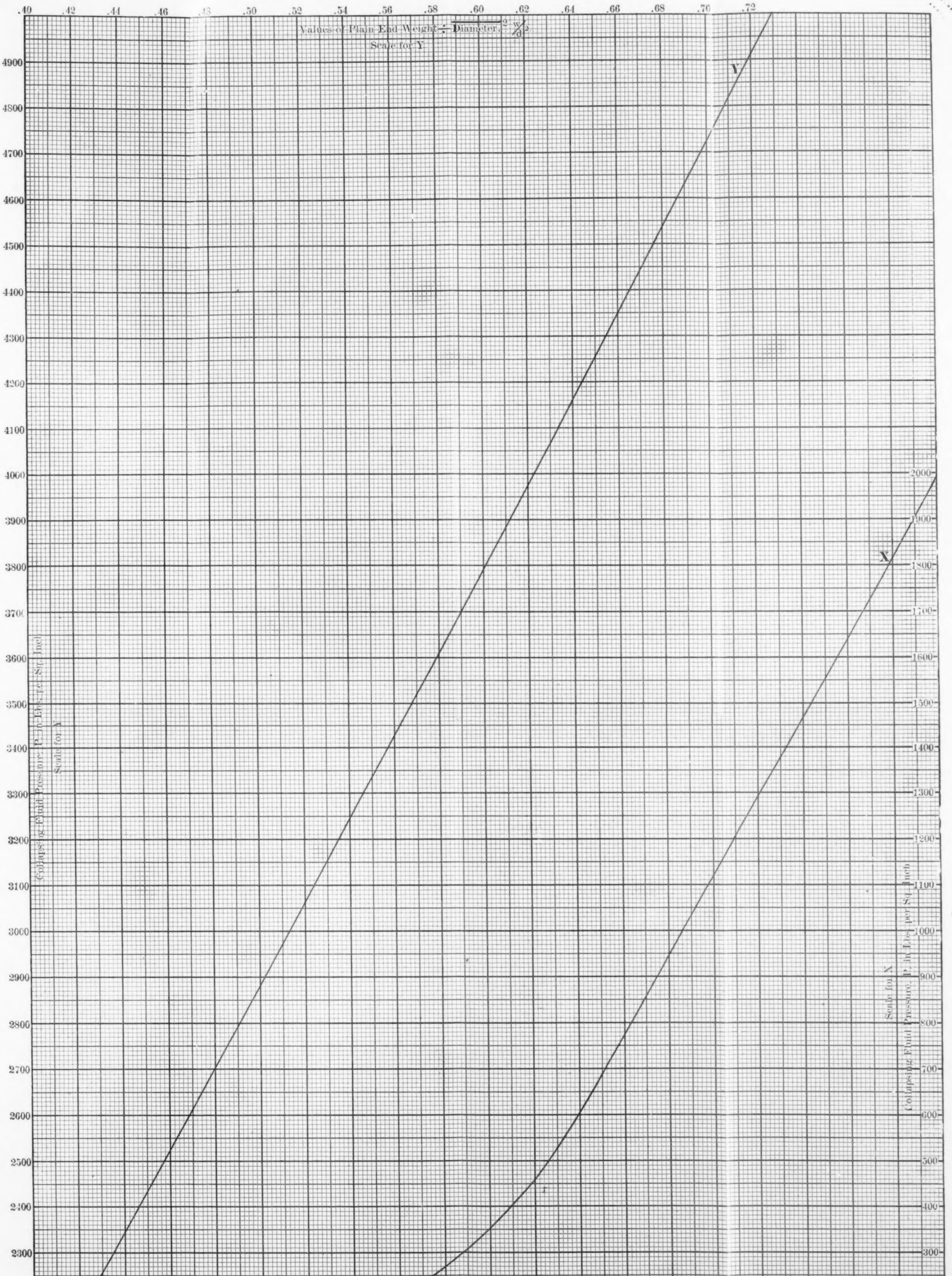


FIG. 61.—CHART FOR OBTAINING COLLAPSING PRESSURES OF LAP-WELDED STEEL TUBES, WHEN PLAIN-END WEIGHT AND OUTSIDE DIAMETER ARE GIVEN. BASED ON PROF. STEWART'S FORMULÆ C AND D. FOR EXPLANATION OF USE, SEE PAGE 819.

Chart Showing Relation of Collapsing Pressure to $\frac{w}{d^2}$. — Fig. 61 resulted from plotting equations C and D (see page 795) to a vertical scale of probable collapsing pressures and a horizontal scale representing the plain-end weight per foot divided by the square of the outside diameter, or the $\frac{w}{d^2}$ contained in the formulæ. The errors of reading this chart should not exceed 5 pounds for the probable collapsing pressure, nor 0.001 for the weight divided by the square of the outside diameter.

This chart is based upon precisely the same experimental data as Fig. 60, the difference being that for any given size of tube this chart shows the relation of the probable collapsing pressure to the plain-end weight, while the preceding chart shows its relation to the thickness of wall. This chart should be used in calculations relating to collapsing pressure when the plain-end weight is either given or required, while the preceding chart should be used when the thickness of wall is given or required.

Example 8. Find the probable collapsing pressure of a $6\frac{5}{8}$ (7 O. D.) inch casing whose plain-end weight is 17 pounds per foot.

Dividing the plain-end weight in pounds per foot by the square of the outside diameter in inches we get $\frac{w}{d^2}$ equal 0.347. Finding this value on the scale at the lower margin of Fig. 61 we look vertically until the line XX is reached, then look horizontally toward the right and read 1,525 pounds per square inch as the probable collapsing pressure required.

While this value is for a 20-foot length of tube, as in the preceding chart, it may be used without substantial error for any length greater than about six diameters, or in this case $3\frac{1}{2}$ feet, between joints tending to hold the tube to a circular form.

Example 9. Find the plain-end weight per foot of a $5\frac{5}{8}$-inch casing (6-inch O. D.) to withstand a maximum difference between external and internal-fluid pressures corresponding to a water head of 1,200 feet, on the basis of a factor of safety of four.

A table of hydrostatic pressures will show that this head of 1,200 feet will create a fluid pressure of 520 pounds per square inch, tending to collapse the casing at its lower end. Multiplying this by the factor of safety we get 2,080 pounds per square inch as the probable collapsing pressure. Finding this value on

the left-hand margin of Fig. 61, we look horizontally toward the right until line YY is reached, then up the nearest vertical line and read 0.41 as the value of $\frac{w}{d^2}$, or plain-end weight divided by the square of the outside diameter. Now since $\frac{w}{d^2}$ equals 0.41, w will equal 0.41 multiplied by the square of the outside diameter, or $0.41 \times 36 = 14.76$ pounds per foot, as the required plain-end weight.

DISCUSSION.

Mr. Henning.—I wish to compliment the author for the character of the paper that he has presented as, in my opinion, it contains valuable information which the Society was not in possession of before. Of course, a great deal of time has been spent on it, and somebody has spent a lot of money on it, but it gives us information which is valuable.

I wish that manufacturers generally would give us as much information about what they produce as has been given here, so that we may know what material will do after it is finished, and not alone what it will do when it is in the shape of a test piece.

Mr. William T. Donnelly.—I would like to ask whether the tube was tested while in the horizontal position or whether it was placed in a vertical position for testing?

Professor Stewart.—These tubes were all tested in a horizontal position. An investigation was made as to what effect the position would have upon the collapsing strength of the tubes, and I satisfied myself that it had no noticeable effect. Many of the tubes were of such weight as to tend to float up, and they would have floated to the top of the test cylinder had they been permitted to do so. This was due to the fact that while under test the tubes were open to the atmosphere on the inside, and were surrounded by water on the outside. Others, of course, tended to sink. In either case the resulting strain was quite insignificant.

Mr. Rice.—As Chairman of the Committee on Papers, I desire to express my personal appreciation to the author of this paper. As far as I remember, it is the most remarkable paper presented at any meeting, and it represents an indescribable amount of work, and I think we should take special notice of it on that account. Then it is notable in respect of the contribution it

makes to the knowledge that we have on the subject. I consider that one of the special functions of the Society, to contribute to human knowledge.

In this connection I want to bring out that many of our members may have knowledge of valuable data or information which, if the request be properly made to the gentlemen who control it, will be permitted to be published for the benefit of the profession.

A Guest.—Some time ago I had occasion to go before the Board of Supervising Inspectors on a subject which is covered very largely by this paper. The occasion for it was that the Board of Supervising Inspectors, at one of their meetings, in their wisdom, had made a new rule designating the thickness of flues in steam boilers coming under the regulations of the marine service, upon a formula which took no cognizance of the length of flue. It was not Clark's formula, but I think some formula that had been adopted by the British Lloyds before the Fairbairn experiments had been measured up. For a year, under the hasty action of the Board of Supervising Inspectors, the condition of the work done for marine practice was quite chaotic, but was finally relieved to some extent by the Secretary of the Treasury suspending the rule. Now I hope that this paper will come to the knowledge of the U. S. Board of Supervising Inspectors, because they need it. They need it now almost as badly as they did at the time I speak of. Through the efforts of some person, whose interests were more largely concerned perhaps than the members of the Board of Supervising Inspectors, a formula was adopted which did take cognizance of the length of flue. Now, I would like to ask for my own information from Professor Stewart, whether he knows from his experiments how they compare with the existing formula of the Board of Supervising Inspectors.

Now as to the matter of the failure of the tubes. The experiments would seem to indicate that the tubes which were tested have perhaps in rolling been laid on a stand, which tended to flatten the tubes at the particular points where they were laid down.

Professor Stewart.—I have not made as yet any such comparison as has been spoken of; but I am satisfied, however, that plain commercial tubes in lengths of about six diameters are no stronger, to any practical extent at least, than similar tubes having lengths up to say 20 feet or more. This is clearly brought out in the

photographs that I have here. I have photographs of all the tubes tested, and if you will look at them you will see that a long tube is only distorted over a small portion of its length. Referring to the photographs of tube No. 50, Fig. 16, p. 757 for example, it will be seen that the left-hand 14 feet of its length has not in any way been distorted by the test. We had a very precise way of determining the distortion which is fully explained in the paper. This photograph of tube No. 50 shows clearly that had we cut off a length of about six diameters from the right-hand end of this tube and put it in the testing apparatus, that this portion of it would have collapsed, just as it did when attached to the 14 feet that showed no distortion whatever.

These commercial tubes, taken at random from the company's stock, were, generally speaking, slightly more out of round near one end than elsewhere along their length. This is clearly shown in the body of the paper (see Fig. 53). The tubes are evidently weakest near one end on the average, but the results of this weakening influence are of no practical importance, not exceeding 13 per cent. and averaging 4 per cent. for a series of determinations made for it.

It is the practice of the National Tube Company, so far as I know, to keep the tubes continually rolling while cooling down, so there is no chance in the regular operation of the mill for a tube to be distorted in the manner suggested by the last speaker.

President Taylor.—Is there any further discussion? If not, I would like to add to what Mr. Rice has said in appreciation of Professor Stewart's paper. It seems to me that the scientific manner in which the subject has been treated is most worthy of commendation. The fact that the tubes experimented with were taken at random from the stock, adds very greatly in my opinion to the value of the tests. It seems to me that we should be very thankful to Professor Stewart and to the National Tube Company not only for making tests of that sort, but for going to the trouble of presenting them to our Society. It is just such papers as this which are of the greatest permanent interest not only to the members of the Society, but to all Engineers the world over, and which gives our Society the international standing which all of us who are ambitious for the Society are anxious to have it attain.

JAMES DREDGE

Honorary Member of the Society

Died August 15, 1906

No. 1117.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

THOMAS RICHARD ALMOND.

Thomas R. Almond was born in Uppingham, Rutlandshire, England, in 1846. He was an inventor of remarkable ability, and as early as twelve years of age was awarded a special prize at a London exhibition for a model working steam engine involving an original and meritorious valve gear.

He came to America in 1866 and established himself as a machinist at Fitchburg, Mass.

In 1875 he removed his business to Brooklyn, N. Y., where it has grown to large proportions developing engineering specialties under the Almond patents. Principal among these are those for the Almond chuck, portable stove lamp, angular shaft coupling, turret head tool, flexible metallic tubing, the club skate, the Almond reaction engine, etc. He has twice received the John Scott medal awarded by the Franklin Institute for meritorious inventions.

He was a charter and life member of the A. S. M. E.; was actively interested in the work of the Society, and contributed numerous articles to the Transactions.

At the time of his death on March 31st, at his home on Dunwiddie Heights, Yonkers, Mr. Almond was a member of the House Committee.

EDMUND BURY.

Mr. Bury was born at Over Darwen, near Blackburn, England, June 5, 1868. He came to this country when a boy, and early in life started work as an apprentice at the shops of the Whipple Mfg. Company at Cleveland, Ohio. This plant later burned down, and he completed his apprenticeship in the Erie shops. He devoted his evenings to studying drafting and mechanics, and after becoming a journeyman worked as a toolmaker, designer and in executive

positions with various tool-building and engineering concerns, among these being Pratt & Whitney Co., Hartford; Fraser & Chalmers, Chicago, and the Ingersoll-Sergeant Drill Co., Easton, Pa. He acquired a broad experience in matters pertaining to the shop and drawing-room and developed into a mechanic of marked ability, becoming in turn a skillful toolmaker, an able designer and a practical and conscientious director of shop operations.

For four years he was at the head of the tool and hardening departments of the Ingersoll-Sergeant shops at Easton, and in that capacity devised many special machines and appliances for the economical production of rock drill and compressor parts. In 1901 he formed a partnership with James Herron of Erie, and a plant was erected at Erie for the manufacture of air compressors known as the Bury Compressor Co., of which Mr. Bury was president up to the time of his death, March 2, 1906.

LOUIS CASSIER.

Mr. Cassier was born at Boston in 1862, and immediately after leaving school began work in that city, in the advertising department of one of the daily newspapers. This he continued until, late in the eighties, he went to New York, and there engaged in miscellaneous advertising work, principally in connection with the American edition of the *London Illustrated News.*

Though not an engineer, engineering appealed to him as a field for publishing exploitation—exploitation of a new kind with hitherto untried methods, and the result, after a comparatively brief period of planning, was the first number of the magazine bearing his name—a magazine of illustrated engineering, intended to deal with steam, electricity, and power. This was in November, 1891 and in the fall of 1894 a London edition of the magazine was started.

Late in 1903 Mr. Cassier purchased *The Electrical Age,* a periodical of many years standing. The Cassier Magazine Company had meanwhile been formed with The Electrical Age Company as a subsidiary organization, and with Mr. Cassier as president of both, and under this new ownership *The Electrical Age* started out on a new career in 1904. An English edition also of *The Electrical Age,* with offices in London, was projected when death overtook him in the railway disaster on the London and Southwestern Railway, on July 1st. Speeding from Plymouth to London, the fast

night express, which carried mails and passengers from the American Line steamship *New York*, left the rails near Salisbury. Twenty-three of the passengers were killed and many others injured.

He was an associate member of the American Society of Mechanical Engineers and of the American Institute of Electrical Engineers, and a member of the Iron and Steel Institute of Great Britain.

GEORGE WILLIAM CATT.

Mr. Catt was born at Davenport, Iowa, March 9, 1860. He received his technical education at Iowa State College of Agriculture and Mechanic Arts, graduating with the degree of B.C.E. in 1882. During the period 1885-1887 he was engineer with Kings Bridge Company, Cleveland; 1887-1893, vice-president and chief engineer San Francisco Bridge Company. While with the latter company he built several large structures on the Pacific Coast, notably a combination three-hinged arch of 340-ft. span across the Fraser River at Lillooet, B. C.

In 1890-91 the Great Northern and other railroads became active in completing their lines in the Northwest, and much of the bridge and trestle work on the lines in the State of Washington was built under Mr. Catt's direction. During this busy period of his career, he constructed in one year fourteen miles of railroad trestle, one-half mile of railroad truss bridges and about one-quarter of a mile of highway bridges.

In 1893 he organized and became president and chief engineer of the New York Dredging Company, which engaged in numerous harbor improvements for the United States Government and for private parties along the Atlantic Gulf coasts, including a ship canal seven miles long at Sabine Pass, Texas. Mr. Catt was largely instrumental in improving machines and methods and making commercially successful the hydraulic system of dredging now in use. In 1899 he resigned and organized the Atlantic, Gulf and Pacific Company, an association of engineers and contractors, of which he was president and chief engineer till his death Oct. 8, 1905.

Mr. Catt was one of the founders, in 1892, of the Northwestern Society of Engineers, at Seattle, and was its first president. He was a member of the American Society of Civil Engineers, and of the Institution of Civil Engineers of Great Britain.

JAMES DREDGE.

Mr. Dredge was born in Bath, England, July 29, 1840. His early training was in bridge engineering with his elder brother William and also with his father.

From 1858 to 1861 Mr. Dredge was in the office of D. K. Clark. In 1862 he entered the office of Sir Jno. Fowler and remained there for three years, being for the most part engaged in the Metropolitan Railway System, side by side with Sir Benj. Baker.

In 1870, upon the death of Zerah Colburn, Mr. Dredge, in association with Mr. William H. Maw and Mr. Alexander T. Hollingsworth took up the publication of *Engineering,* which Mr. Colburn had started in 1866, and it was in this publication his first writings appeared.

In addition to the editing of *Engineering,* Mr. Dredge and Mr. Maw jointly wrote "Road and Railway Bridges," and in 1873 issued the "Report of the Vienna Exposition." Some of Mr. Dredge's later works are "History of the Pennsylvania Railroad," in 1877; "Electric Illuminations," 1881; "Report of the Paris Exposition," 1889, to which he was Commissioner; and, "Modern French Artillery," 1891, for which he received a second decoration from the French government. He was also the author of an article on "Gas Lighting by Incandescence," and an article on Henry Bessemer, published in "Transactions."

The American Society of Mechanical Engineers made him an Honorary Member in 1886. In 1890 he came to America and visited Chicago. In the spring of 1891 the Queen appointed the Royal Commission, consisting of the Council of the Society of Arts, of which Mr. Dredge was a member, and in September of that year he came to this country as a Royal Commissioner, accompanied by Sir Henry Wood, secretary of the Commission. They went to Chicago and made a joint report, which resulted in new and more energetic measures in England to promote the British exhibit there. The first paper was read by Mr. Dredge to the Society of Arts, December, 1890, and resulted in the British Commission. The second paper was read to the same body December 9, 1891. These were followed by one read January 12, 1892 to the London Polytechnic.

Mr. Dredge was a member of the Savage Club, also of the St. Stephens and one of the Council of the Society of Arts and was also a member of the Scandinavian Club.

WILLIAM R. FLEMING.

William R. Fleming was born in Harrisburg, Pa., May 9, 1862, and spent most of his life in that city.

After eighteen months in the works of the Harrisburg Foundry and Machine Company, he went to the Pratt and Whitney Company in Hartford, Conn., where he learned the practical side of mechanical engineering.

After four years Mr. Fleming removed to New York and represented the Harrisburg Foundry and Machine Company, and in 1898 he was made manager of the company, and remained there until within a few months of his death.

During his service he made many improvements, and took out patents both on the electrical as well as the mechanical features of the Harrisburg Company's manufacture.

Mr. Fleming was a member of many technical societies and founder of the Engineers' Club of Central Pennsylvania.

He died suddenly of heart trouble in Washington, D. C., June 6, 1906.

GEORGE A. GRAY.

George A. Gray, the founder and for many years president of the G. A. Gray Company, Cincinnati, Ohio, died at his home in that city, on June 14, 1905. Mr. Gray was born in 1839 on a farm in Illinois, and his boyhood was spent there. Later he came to Cincinnati and served an apprenticeship with Miles Greenwood, devoting his attention to the study of mechanics. In about 1866, with Messrs. Gordon and Gaff, he bought the machine tool department of the old "Niles Works," which had been started by a man named Niles, calling the new firm the Niles Tool Works. After continuing in that location for several years the plant was removed to Hamilton, Ohio. Later Mr. Gray sold his interests in the firm and started the manufacture of planers in Cincinnati, in which business he continued until last May, when he retired from active management of the company which bears his name.

Mr. Gray's mechanical experience covered a wide field. As a young man during the Civil War he was engaged in building monitors for the United States Government, and about the same time he invented a multiple firing gun.

FREDERICK GRINNELL.

Frederick Grinnell was born at New Bedford, Mass., August 14, 1836, and died in his home town October 21, 1905.

In 1852, after attending the Friends' School in New Bedford, he entered the household of his uncle, Joseph Grinnell, of New York, where he prepared for Rensselaer Polytechnic Institute, in which he completed the four years' course in three years, graduating in 1855 at the head of a class of sixty students. After graduation he became connected with the Jersey City Locomotive Works, and later became the mechanical superintendent of the Atlantic and Great Western Railway, in which position he built over one hundred locomotives.

In 1860 he became treasurer and superintendent of the Corliss Steame Engine Works at Providence, and five years later returned to the Jersey City Locomotive Works as general manager.

In 1869 he purchased a controlling interest in the Providence Steam and Gas Pipe Company, and about eight years later obtained the license to manufacture the Parmelee Automatic Sprinkler, which he improved, and introduced with great energy.

In 1881 he invented the automatic sprinkler which bears his name.

In 1892 he combined the business of his sprinkler with that of the manufacturers of leading devices of the character and retained the management of the whole business until his retirement shortly before his death.

ISAAC V. HOLMES.

Isaac V. Holmes, a charter member of the Society, was born Aug. 9, 1835, in Jersey City, N. J., and died suddenly of acute heart disease April 28, 1906, at Wheaton, Ill. He was educated at Fulton Academy, Fulton, N. Y., and in 1849 became apprenticed to the Novelty Iron Works in New York City. After serving his time as apprentice he continued in their employ as draftsman until 1853, when he was given charge of the designing and erection of machinery for operating iron mines in the neighborhood of Port Henry, near Lake Champlain.

At the completion of this work he returned to the Novelty Works and remained with them as a designer, and later as superintendent, until the works were closed in 1869. It was under his supervision that the *Monitor* was constructed.

Mr. Holmes was appointed a member of the United States Commission for the investigation of boiler explosions, of which Professor Thurston was chairman. From 1869 to 1873 he was engineer and superintendent of the John Casper Co., Mt. Vernon, Ohio.

From that year he devoted his time to expert engineering work, with offices at Cleveland, Ohio, and later in Chicago. His principal line of work was the design and construction of factory and power plants, and patent expert work.

For the last eight years Mr. Holmes had devoted a large share of his time to the experimental development and final design of a new system of feed water purification, which he was bringing into commercial value at the time of his death.

JOHN CHRISTIAN KAFER.

Passed Assistant Engineer John Christian Kafer, U. S. N., retired, died in Trenton, N. J., March 30. He was born in Trenton, N. J., Dec. 27, 1842. The outbreak of the Civil War found Mr. Kafer in the prosecution of engineering studies and work, and in January, 1863, he was appointed third assistant engineer in the Navy. On the U. S. S. *Mackinaw* he served through the campaign on the James River and in the first attack on Fort Fisher.

On the morning of President Lincoln's death Mr. Kafer sailed for the Mediterranean on the old U. S. S. *Kearsarge,* whose executive officer was Lieutenant-Commander George Dewey. During the term of Commodore Loring as engineer-in-chief of the Navy, Mr. Kafer, an old shipmate, became his principal assistant and served in a similar capacity under Engineer-in-Chief Melville. He taught at the Naval Academy from 1868 to 1874 and from 1875 to 1882, and was retired June 18, 1888, for disability incident to the service. In 1885 he declined the professorship of mechanical engineering at Cornell University, and a little later he became associated with the Morgan Iron Works of New York City, serving as vice-president, superintendent, secretary and treasurer. He then became connected with the Quintard Iron Works, and within the last month had organized the consulting firm of Kafer, Mattice and Warren. Mr. Kafer was one of the most active members of the Engineers' Club, was president from 1901 to 1904, and had been for many years one of the Board of Governors. He also served as a member of the Council of the

American Society of Mechanical Engineers, of which he had also been vice-president. He was the senior American member of the Institution of Naval Architects of Great Britain, a member of the American Society of Naval Engineers, and the Society of Naval Architects and Marine Engineers. He was treasurer of the Building Committee of the United Engineering Building at the time of his death.

JAMES R. F. KELLY.

James R. F. Kelly was born in Greonock, Scotland, on March 14, 1844.

He was an apprentice in the machine shop at the Greonock Foundry Co. for two years, and for three years in drawing room.

From 1865-1868 he was with the Novelty Iron Works of New York City in their drawing room, and was draftsman and mechanical engineer for the U. S. Engineer Corps, in charge of the improvement of the East River and Hell Gate for ten years, and with the U. S. Electric Lighting Company of New York 1878-1880.

From 1882-1894 Mr. Kelly was connected with the firm of Joseph Edwards & Co., engineers and machinists of New York City, as partner and general manager.

At the time of joining the Society, in June, 1895, and up to 1897 he was connected with the firm of James R. F. Kelly & Co. From 1897 to May 1, 1905, he was associated with Mr. W. D. Kelley. Mem. Am. Soc. C. E., in the firm of Kelly and Kelley, Engineers and Contractors.

May 1, 1905, to the time of his death, on December 11, 1905, he was president of Kelly and Kelley (Incorporated), Engineers, Builders and Contractors, 45 E. Forty-second Street, New York City.

ALBERT P. LOSCHER.

Albert P. Loscher left Leipsic, Germany, when under sixteen. He travelled through Switzerland, Belgium and France, finally reaching London, England, before he was seventeen. He became manager of A. Pittler's showroom, High Holborn, before he was twenty. (This Pittler was patentee of automatic lathes, which bear his name.) In 1902 he became works manager to W. H. Bailey & Co. (Ltd.), Hydraulic Engines, Manchester; resigned on account of his health. He then took a position as tool expert with The National Gas Engine Co., but his

health caused him to resign again, and journey farther West to get relief. His new position was as feed and speed expert, Westinghouse Machine Co., East Pittsburgh. He resigned May, 1904, and went to Denver for some months. On returning to Pittsburgh he acted for some time as tool specialist to The H. K. Porter Co., but resigned from their service February, 1905, and left for Los Angeles March 1, 1905. Here he formed a partnership to manufacture gasoline engines. He died June 11, 1905.

ALEXANDER CAMPBELL McCALLUM.

Alexander Campbell McCallum died September 23, 1905, at Pasadena, Cal. He was born in Glasgow, Scotland, May 16, 1867. He began his mechanical work as an apprentice in 1881 with W. B. Thompson Co., shipbuilders in his native city. He secured drawing room and further shop experience with the Chas. Connal Co., Fairfield Shipbuilding Co., Singer Sewing Machine Co. and the Hyde Park Locomotive Works. He spent the year 1887 at sea on board a City Line steamer from Glasgow to Calcutta. He received the diploma in Naval Architecture from Kensington Schools, London, England. In May, 1888, he emigrated to Canada and entered the employ of the Wm. Hamilton Mfg. Co. of Peterborough as draftsman, and remained with that concern for fourteen years as their works engineer, then in charge of their drafting department, and in charge of erection of outside work since 1893. A few months before his death he entered the employ of the Canada Foundry Co., in charge of the drafting room, and later became assistant superintendent. In 1888-1892 he was instructor in Mechanical Engineering, Mechanics' Institute of Peterborough. He was an Associated Member of the Canadian Society of Civil Engineers.

FRANK S. MEAD.

Mr. Mead was born May 17, 1845, at Tiverton Four Corners, R. I. He was educated at the old Lyons school on College Hill in Providence. He enlisted with the Tenth Rhode Island Volunteers and served until the regiment was mustered out.

Returning to Providence, he entered into business with his father, Lewis P. Mead, in the Cove Foundry and Machine Company and L. P. Mead & Co. After being associated with his father a few years he engaged in the stove business in Omaha, Neb. He re-

turned to Providence in the early '80s, when insulated wire was in its infancy, inventing a number of machines for this line of work.

He became associated with the late Eugene F. Phillips, the founder of the present American Electrical Works, and was the manager of the Eugene F. Phillips Electrical Works in Montreal, Canada, for a period of ten years.

He returned to Providence for the manufacture of the Mead gas engine, but did not continue in this line of business long, as he re-entered the employ of Mr. Phillips. At the time of his death, May 18, 1906, Mr. Mead was superintendent of the Auburn plant of the Washburn Wire Company.

J. VAUGHAN MERRICK.

J. Vaughan Merrick, born August 30, 1828, in Philadelphia, died March 28, 1906, was the son of Samuel Vaughan Merrick, who was the founder of the Southwark Foundry and Machine Co., The Franklin Institute, and the first president of the Pennsylvania Railroad Co.

After his graduation from the Philadelphia High School he entered the works of Merrick & Towne (the original firm name of the "Southwark Foundry") as an apprentice, and went through the various mechanical departments. In 1849 he became the head of the firm of Merrick & Sons, the successors of Merrick & Towne.

For the following twenty years his energies were devoted to the development of the "Southwark Foundry." During the Civil War the Foundry did work for the Government, and Mr. Merrick devoted his personal services to this part of the business.

He was selected by the Navy Department in 1862 as a member of the board of experts to report on naval machinery.

He designed and built much of the machinery for the following naval vessels: *San Jacinto, Wabash, Yantic, New Ironsides, Wyoming* and *Yazoo*. The new *Ironsides* was one of the first of our ironclads. Until his death Mr. Merrick was a member of the board of the company, the title of which is now "Southwark Foundry and Machine Co."

In 1833 Mr. Merrick was appointed a member of the board of experts to look into the best manner of improving the water supply for Philadelphia.

In 1870, owing to impaired health, he retired from active business and devoted his remaining years largely to philanthropic work.

In 1873 he was instrumental in founding St. Timothy's Working Men's Club and Institute, Roxborough, which still exists.

He and his wife founded St. Timothy's Hospital and House of Mercy, giving the land and its original buildings, together with a substantial endowment for its support, and he labored hard and continuously for it until his death.

The following list of positions filled by Mr. Merrick will give some idea of his energy:

Trustee of the Episcopal Academy from 1874 to 1898.

Trustee of the University of Pennsylvania since 1870, and was senior trustee at his death. Member of Committee on University and Chairman of the Standing Committee on "Department of College and Philosophy."

Member of Board of Managers of the Episcopal Hospital and Chairman of the Building Committee from 1876 to 1900.

Incorporator of the "Wagner Free Institute of Science," in 1864. Trustee from 1885 to 1894.

Director of the Zoölogical Society since its origin, and Vice-President since 1886.

For over thirty years Director of the "Society for the Relief of Widows and Orphans of Deceased Clergy of the P. E. Church."

One of the founders of the "Free and Open Church Associations," and President of same since its origin in 1873.

At the University Day exercises, held on February 22d, 1906, the Trustees of the University of Pennsylvania conferred the honorary degree of Doctor of Science upon Mr. Merrick.

He was also a member of the Union League, Penn and Philadelphia Clubs of Philadelphia.

Mr. Merrick joined the Society in May, 1881, and was Vice-President in 1883-1885.

MAX HOWARD MINER.

Max H. Miner, a member of the editorial staff of *The Railway Age*, was taken ill suddenly on October 31 while at his desk in the New York Office, and died at noon Tuesday, November 7th, 1905. Mr. Miner was twenty-nine years of age, having been born at Charlemont, Mass., on August 25, 1876. After the usual high school course he attended Cornell University, and graduated from Sibley College with the class of 1899. He served as special apprentice for one year in the shops of the Illinois Central R. R. at

Chicago, after which he returned in January, 1901, to Cornell as Instructor in Experimental Engineering. In December, 1901, he joined the staff of *The Railway Age*.

CHARLES C. NEWTON.

Charles C. Newton, President and Treasurer of the Newton Machine Tool Works (Incor.), Philadelphia, died at Bremen, Germany, on June 13, 1906. He had gone to Europe to seek relief at the baths at Bad-Nauheim, Germany, but was too ill to continue his journey after his arrival at Bremen, nine days previous.

Mr. Newton was born February 9, 1846, at Cambridge, Washington County, N. Y., and at the age of nineteen he indentured himself as apprentice to the Brooks Locomotive Works, at Dunkirk, N. Y., spending most of his time in the tool room, where the foundation of his successful after career was laid. After serving his apprenticeship, and working in various shops, Mr. Newton returned to Dunkirk in 1875, and, in partnership with Mr. J. D. Cox, manufactured twist drills, reamers, cutters, etc., removing to Cleveland, Ohio, in 1876, to broaden the scope of their work, which had largely increased, operating under the firm name of Newton and Cox. In 1880 Mr. Newton sold out his interest to Mr. Cox, who formed out of the old firm the Cleveland Twist Drill Company.

Removing to Philadelphia, Mr. Newton laid the foundation for the present Newton Machine Tool Works (Incor.), in a little shop at No. 2341 Callowhill Street, with only himself and an assistant as the working force. In this shop he designed the first heavy railroad tools. As the demand grew for his various designs, he was forced to seek larger quarters. In the fall of 1886, he built a new shop at the corner of Twenty-fourth and Wood Streets, and there designed machines for sawing metal cold, and also a large number of new tools, for which he took out patents. Mr. Newton built in August, 1895, the first part of the present Shop on the corner of Twenty-fourth and Vine Streets, which has since been increased to take in the entire square bounded by Twenty-third, Twenty-fourth, Vine and Wood Streets.

Mr. Newton was the sole proprietor of the works until July 14, 1897, when articles of incorporation were taken out. The cor-

poration title was the Newton Machine Tool Works (Incor.), with Mr. Newton as president and treasurer.

HIRAM PEARSON.

Hiram Pearson was born in Scranton, Pa., September 22, 1862, and after passing through the common schools of that city was apprenticed to the Lackawanna Iron and Steel Company at Scranton, from 1879 to 1883.

For the next five years he was in the service of the Dickson Manufacturing Company, and from 1888 to 1894 was general foreman of the Boies Steel Wheel Company; from the latter date to the time of his death, May 4, 1906, he was with the General Electric Company, passing through the drawing room and office of operating engineer to the position of chief engineer of the power plant of the Schenectady Works.

Mr. Pearson was a member of a number of fraternal organizations, and was buried with Masonic honors.

ALBERT J. PITKIN.

Albert J. Pitkin was born at Northampton, Ohio, in 1854, and died at his home, in New York City, on November 16, 1905. At the age of seventeen he entered the apprenticeship of the Stationary Engine Works of the Webster, Camp and Lane Machine Company of Akron, Ohio. He then spent a year in the locomotive repair shops of the Cleveland, Akron and Columbus Railroad, after which he entered the drawing office of the Baldwin Locomotive Works. After five years at the Baldwin Works he became chief draftsman of the Rhode Island Locomotive Works, and two years later, in 1882, was appointed mechanical engineer of the Schenectady Locomotive Works. Two years later he became superintendent. Upon the death of the president, Mr. Edward Ellis, Mr. Pitkin was made vice-president and general manager, and from that time developed the commercial as well as the manufacturing features of the business which gave these works their high standing among the locomotive building companies of the world. Upon the formation of the American Locomotive Company, Mr. Pitkin became its first vice-president, and upon the death of Mr. Callaway in June, 1904, Mr. Pitkin was elected president. He was a most generous, considerate employer, and was sincerely interested in the welfare of the great army of men under him. He

realized the vital relationship between the locomotive and human welfare, and one of his greatest pleasures was found in his devotion to its development and improvement.

Mr. Pitkin had been a member of the Society since 1882.

HERMAN POOLE.

Herman Poole was born in Roxbury, Mass., March 9, 1849. He was graduated from the public schools of Boston, and at the age of sixteen entered the Massachusetts Institute of Technology, where he remained a year. He also passed one year at Cornell University.

In 1869 and 1870 he served an apprenticeship with Kent and Williams, and in 1885 gained a shop experience with Stevenson's Boiler Works, Ontario, and was also chief chemist and in charge of construction of the Alpha Oil Gas and Mining Company, and during 1887-1891 chief chemist of the Casselle Chemical Company of Cleveland. Later Mr. Poole was in charge of construction of the plant of the International Phosphate Company, and from then to the time of his death, February 6, 1906, in general chemical work and construction.

He was the author of several works on calorimetry, one of his best known works being "Calorific Value of Fuels."

He was a member of the American Institute of Mining Engineers, American Chemical Society and the Electro-Chemical Society.

JOHN FRANKLIN SEAVEY.

Mr. Seavey was born in Charlestown, Mass., on September 27, 1865.

His early education was secured in that city and after graduation from the Charlestown High School, entered the Massachusetts Institute of Technology from which he received his degree in 1886.

Immediately afterward he entered the City Engineer's office of the City of Lowell, Mass., and remained five years, 1886-1891, then four years as mechanical engineer with the Ludlow Manufacturing Company; then a year as inspector with the New York Mutual Fire Insurance Company; then for two years with George S. Rice and George E. Evans, civil and hydraulic engineers, and for five years with the American Steel and Wire Company of Worcester, Mass.

At the time of his death, January 14, 1906, in South Bethlehem, Mr. Seavey was assistant chief engineer of the Bethlehem Steel Company.

T. JACKSON SHAW.

T. Jackson Shaw died July 13, 1905, and was buried at Wilmington, Del. He was born at Wilmington on July 24, 1852. His education was received in the high school at Norristown, Pa., after which he served an apprenticeship at the pattern making trade in the marine engine building shops of Wood, Dialogue & Co., Camden, N. J., after which he was transferred to the machine shop, and then to the drafting room. His work consisted of the design of marine engines, machinery, etc., as well as steamship hull construction.

At the time of joining the Society, at its Nashville meeting, in May, 1888, Mr. Shaw was General Superintendent with the Harlan and Hollingsworth Company of Wilmington, Del.

He went to Wilmington in 1876, and in that year designed the first compound engine constructed by the Harlan and Hollingsworth Company, being for the steamship *Decatur H. Miller*, of the Merchants' and Miners' Line. Mr. Shaw was a director of the above company for a number of years, and in 1901 was elected vice-president, which office he held until February, 1905, when he resigned on account of continued ill-health.

GEORGE V. SLOAT.

George V. Sloat died February 14, 1906. He was born on January 11, 1827. Entering upon his apprenticeship in 1844 in the Morgan Iron Works, New York City, he remained there six years, and then became second assistant engineer on the steamer *New Orleans*. After two years he returned to the Morgan Iron Works. Later he became chief engineer of the *Roanoke*, which ran between New York and Richmond, and was afterwards transferred as chief engineer to the *Jamestown* of the same line.

At the breaking out of the Civil War the *Jamestown* was seized by the authorities at Richmond and Mr. Sloat with the crew came to New York, where he entered the government service and was detailed as engineer to the United States gunboat *Augusta*. He saw considerable service at Hampton Roads and Port Royal, and took part in the capture of Hilton Head. Temporary ill-health

made it necessary for Mr. Sloat to resign his commission. After his recovery he became engineer of the steamship *Golden Rule*.

Immediately after the close of the war Mr. Sloat entered the service of the Old Dominion Steamship Company, New York, and was chief engineer of a number of the earlier vessels of this line running between New York and Richmond. During the time that he was employed in this capacity he gained the experience as superintending engineer of the Old Dominion Company, which position he assumed in 1871, and from this date up to his retirement from the company's service fifteen seagoing steamers and twenty inland steamers were built for the company under Mr. Sloat's direct supervision.

At the age of 73 Mr. Sloat resigned his position with the Old Dominion Company and retired from active professional life.

He had been a member of our Society since 1892.

THOMAS GARDNER SMITH.

Thomas Gardner Smith was born in Cincinnati, Ohio, on the 19th of March, 1862. He was educated at the public schools of Cincinnati and received the degree of Mechanical Engineer at the Stevens Institute of Technology, in 1885. He was in the shop of the Indianapolis Division of the Pennsylvania Railroad the following year, and then with Henry Warden in Philadelphia; Gorden Strobel and Laureau, Philadelphia; C. R. Vincent & Co., New York; Ball and Wood, New Jersey; Pennsylvania Railroad at Indianapolis, and from 1892 to the time of his death, the 20th of November, 1905, was a consulting engineer of the city of Cincinnati.

JOHN STANTON.

John Stanton, one of the most conspicuous men in the copper industry in the United States, died at his residence, New York, on February 23, 1906, after a brief illness. Mr. Stanton was born in Bristol, England, February 25, 1830.

In 1849 he took the position of clerk and later of assistant manager of the "Swedes Mine" and the Mt. Pleasant Mine near Dorr, N. J. In 1852 he took up the business of exploring for and developing copper deposits, taking direction in the field of a variety of operations. In 1854 he bought for a New York company the Eureka Mine at Ducktown, Tenn., taking charge as general manager. He erected its equipment, including a complete smelting

plant of reverberating and cupola furnaces; and also a plant for treating the oxidized ores and recovering their copper contents in the form of cement copper. He assumed the general management of the following companies: The Central Mining Co., in 1864; the Atlantic Mining Co., in 1872; the Albany Mining Co., in 1884, and the Wolvering Mining Co., in 1890.

He was a director in three companies as well as the Baltic Mining Co., the Locke Drill Co., the Michigan Copper Mining Company, the Mohawk Mining Co., Phœnix Consolidated Copper Co., and the Winona Copper Co. An approximate estimate of the copper produced from the various properties in which he was interested in an official capacity, at the time of his death, would total close to $90,000,000 per annum.

On the formation of the Copper Producers' Association of the United States in 1892, which organization embraced the representatives of all the principal copper mines, Mr. Stanton was chosen by common consent as its executive officer and statistician, holding the position until the abandonment of the organization.

He was one of the founders of the New York Mining Stock Exchange in 1876 and for two years was its president. After that he became its treasurer, which position he held until his death. He was also one of the founders, and for two years president, of the Engineers' Club. He was a member of the American Institute of Mining Engineers, of the Lake Superior Mining Institute and of the North of England Institute of Mining and Mechanical Engineers. Mr. Stanton joined the Society in June, 1891.

JOHN EDWARD STEVENS.

John Edward Stevens was born in St. Petersburg, Russia, in 1846. He was educated at Croydon Academy, England, and the Mechanics' Institute, Leeds. From 1861 to 1868 he was apprenticed at machinery construction with P. Fairbanks & Company, Leeds. Towards the completion of his apprenticeship he had charge of erecting gangs and erection of extensive systems of machinery and mills. From 1872-1876 he was engineer and manager of Narva Flax Mills in Russia. He here prepared complete plans for new mills, including the installation of water power; also complete plans for the iron construction for execution in Great Britain. From 1876 to 1881 was travelling representative for Fairbairn, Kenned and Naylor, machinists, of

Leeds, travelling for them in Germany, Italy, Austria, Russia and the United States. During 1881-1887 he was superintendent of Ludlow Manufacturing Co., Ludlow, Mass.

Mr. Stevens joined the Society in 1900.

FREDERICK TALLMADGE TOWNE.

Frederick Tallmadge Towne, Member of the Society, died at Stamford, Connecticut, on February 4, 1906.

Mr. Towne was born at Stamford, Conn., on March 5, 1872, being the second son of Henry R. Towne, past president of the Society; and during the comparatively short life which was given to him he attained eminence in the engineering profession.

After a preliminary education at St. Marks' School, Southboro, Mass., Mr. Towne went to the Massachusetts Institute of Technology, at Boston, from which institution he was graduated in 1892, after which he entered the works of the Yale and Towne Manufacturing Company, beginning in a subordinate position, and passing through the various departments of the works in order to obtain a practical familiarity with the multifarious operations connected with the manufacture of locks and hardware. In 1896 he was appointed assistant to the president, and at the close of 1898 he became general superintendent of the works, which position he held until the time of his death.

During this time the works grew from an establishment employing twelve hundred men to one giving occupation to more than double that number, while under his leadership there were introduced into the establishment methods and systems representing the latest developments of manufacturing and industrial science. His talents for organization and systematization showed themselves to be of the highest order, while with the introduction of such methods there grew up also relations of deep affection and association with practically everyone under him in the entire establishment.

Apart from his duties in connection with the internal administration of the works of the Yale and Towne Manufacturing Company, Mr. Towne took an active interest in external affairs. In 1900 he was chosen a member of the Advisory Council of the National Founders' Association, of which society he became president in 1903. He was active in organizing the Manufacturers' Association of Stamford, and was also a member of a number of social and professional organizations.

Although he had suffered serious illness at various times, he was not supposed to be in especial ill-health, when, on February 3, 1906, being Saturday afternoon, he presided at the meeting arranged for the award of prizes to the various employees of the company for useful suggestions in connection with the work of the establishment. He made an effective address, appropriate to the occasion, presented the prizes, and closed with the words: "Good night!" As he resumed his seat he fainted, and was carried away to his home, suffering from an acute attack of Bright's disease of the kidneys, from which he never recovered consciousness, passing away early on the following morning, February 4, 1906, in the thirty-fourth year of his age.

Mr. Towne became a Junior Member of the Society on June 26, 1895, and was promoted to full membership on May 28, 1902.

MATTHEW PATTERSON WOOD.

Matthew Patterson Wood was born January 23, 1835, in Adams, Mass. After receiving his education, he was apprenticed as a machinist, 1854-56, in the Lawrence Locomotive and Machine Shop, Lawrence, Mass. In 1857-58 he was employed as draughtsman and foreman of locomotive erecting shop of the Michigan Southern and Northern Indiana R. R. at Adrian, Mich. During 1859-61 he was master mechanic of the Louisville, New Albany and Chicago R. R.

During the Civil War he was superintendent of motive power of the United States Military Railways, Department of Virginia, Tenn., serving under General Herman Haupt. He was also a confidential agent for Secretary Stanton. From 1865 to 1867 he was superintendent of the Ohio Falls Car and Locomotive Co. During the twenty years between 1870 and 1890 he was principally engaged in railway construction in different parts of the country. He then became superintendent of the Douglas Axe Mfg. Co., at East Douglass, Mass. The last few years of his life he devoted to a general consulting practice. He joined the Society in May, 1890.

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